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Heating · Piping and Air Conditioning

February, 1930

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Air Conditioning—
By EDGAR S. MILLER
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Journal Section
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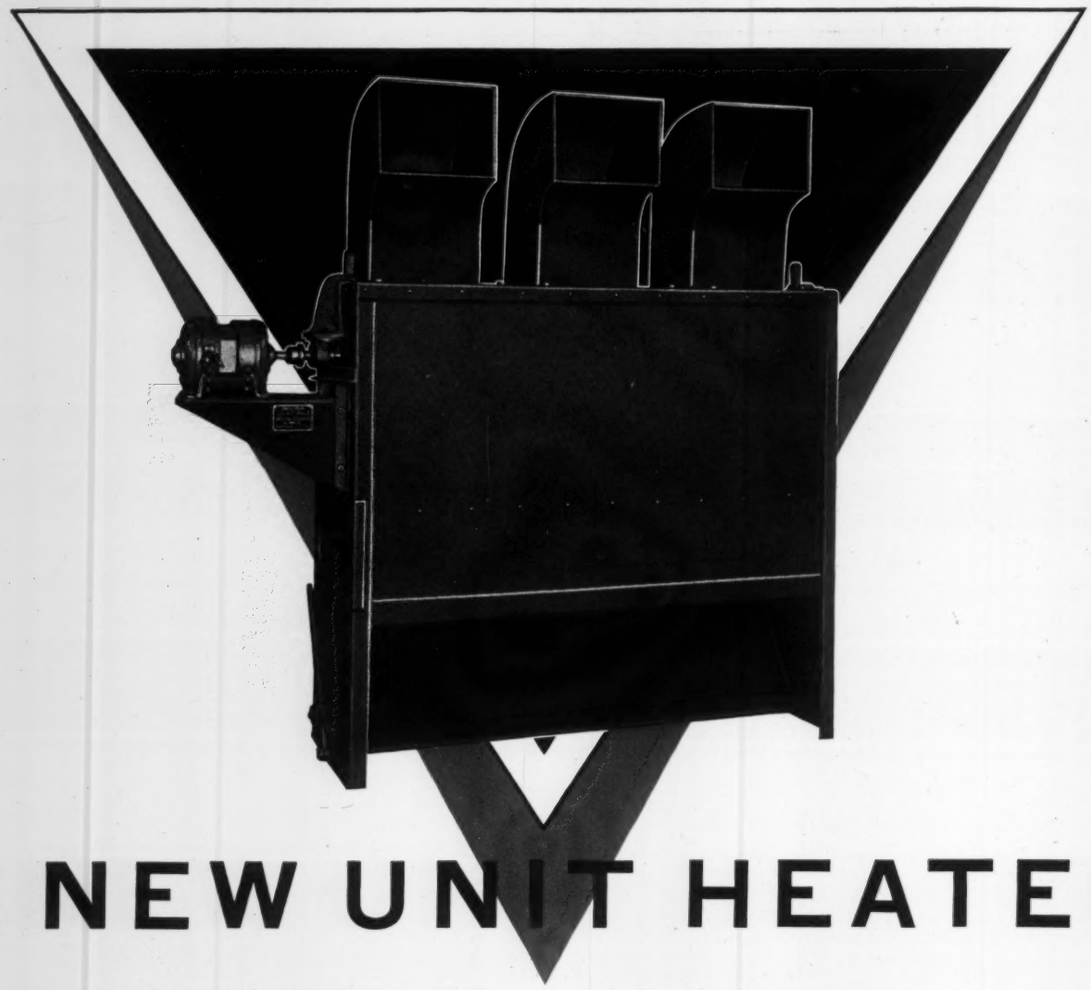

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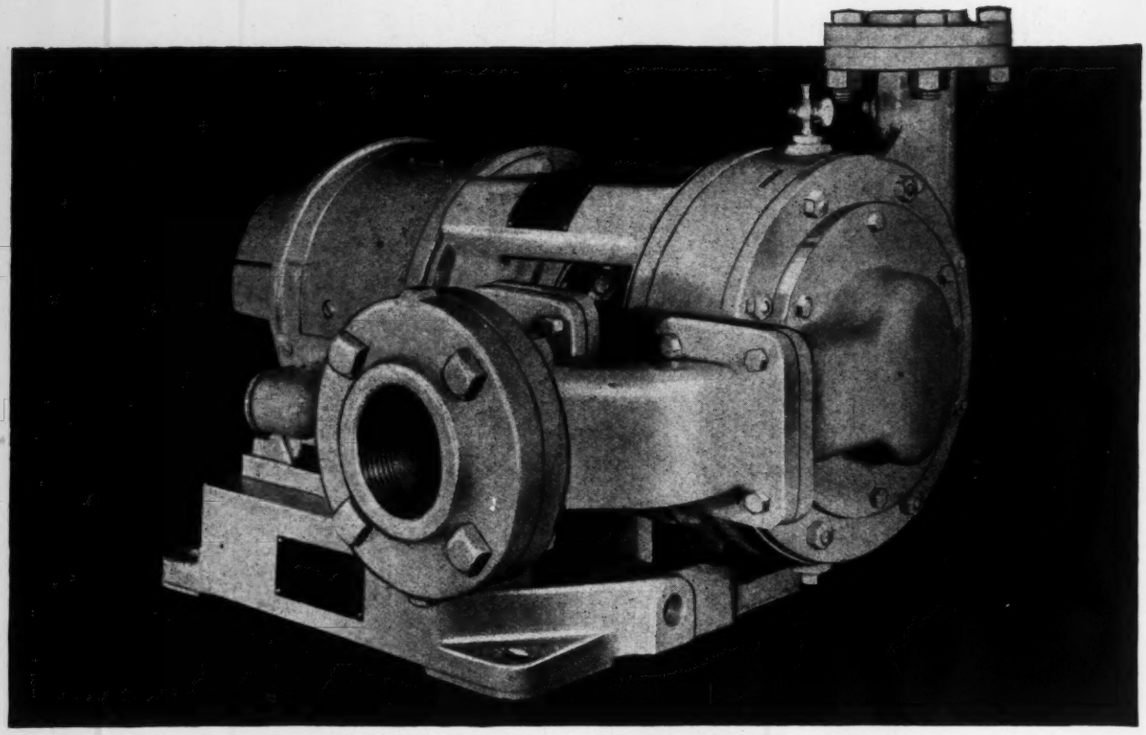

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Heating · Piping and Air Conditioning

Vol. II February, 1930 No. 2

Heat Balance and Toasting "Luckies"

With a Description of the Piping

By A. W. Moulder

HOW much can we make by scrapping the old power plant? Sometimes it is called a steam generating plant or a heating boiler plant; no matter what the cognomen the question is the same. Consulting engineers are busy today finding the answer for many industrial concerns. That answer can be as definite and specific as the answer the insurance man gives to the owner who is seeking to reduce his insurance expense by installing an automatic sprinkler system.

The problem is to determine and set down by measurement or calculation on the one side the steam requirements for heating, processing, etc., and on the other side the electrical requirements. From this is obtained what, in a broad way, is called a "heat balance" for the plant. Balance is a good word because the final answer from which the executive reaches his decision is naturally presented in dollars and cents.

To arrive at that, the consulting engineer carefully prices:

A. The present cost of buying or producing the required amount of heat or process steam, plus the cost of buying or producing the required amount of electrical power. (These items are separated because of the varying length of time used each year.)

B. The estimated cost of producing heat, process steam and power under the proposed new plan. In this, of course, careful attention is given to all new processes and improvements rendered possible by changed conditions.

The balance between *A* and *B* will show either a profit or a loss. The amount of that profit weighed against the cost of investment in new equipment is the "trial balance" from which the executive makes his decision.

Piping for High Pressures and Temperatures

This problem is receiving renewed consideration today as compared with a few years ago because the development of apparatus and piping for higher pressures and temperatures has changed its entire complexion. It is now practical to start with steam at the throttle valve of a turbine at a pressure and temperature sufficiently high so that steam may be extracted at correspondingly high pressures and temperatures, and this steam may be delivered readily anywhere in the plant. Obviously this allows the use of steam (which has already generated much power) for many processes, such as drying, cooking, etc., which formerly required live steam right from the old boilers. Higher temperatures have also made possible the utilization of steam for many processes where previously steam was rarely considered.

Not the least change in condition which may be given consideration in this problem is in the piping. Many of us still think of exhaust steam as something which very closely resembles that white stuff which puffed out of the exhaust head at the old power plant. Also that it is something to be carried in big pipes, because the energy is pretty well taken out of it and it's something that can't be carried very far away from its source. We really know better but I believe many of us (not such old timers either) still have that old picture in the back of our minds.

Of course, it's all wrong, because today we are extracting steam from the turbines at almost any practical pressure and temperature to fit our needs and distributing it all over the plant through relatively small piping made possible by the higher pressure of the steam itself.

This change in piping starts right at the boilers. It is true that for these higher pressures, heavier and better materials are required; better and stronger pipe joints are necessary; and with infinitely more care demanded in the design of the piping and in its proper support. Provision must also be made for greater expansion and contraction. A portion of this extra cost for heavier and better material at least is compensated for in the smaller piping necessitated.

After the steam is extracted from the turbines, the chances are the pressure will still be high enough to allow distribution with minimum sizes of piping and, of course,

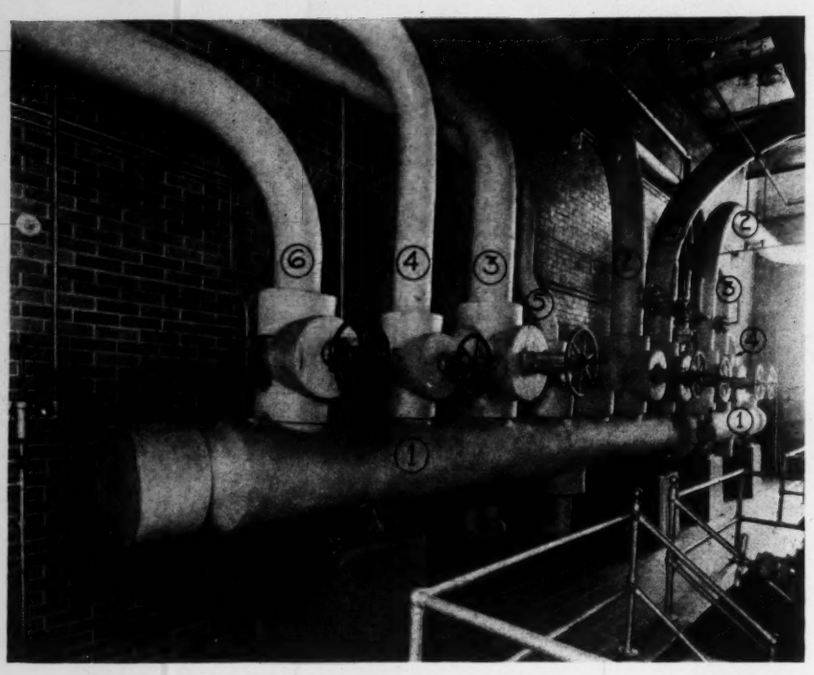


FIG. 1—SHOWING THE 12-INCH MAIN HEADER, OF WELDED CONSTRUCTION

with quality and weight of piping materials consistent with long-life expectancy for the pressures and temperatures involved.

When a large industrial concern decides to put in a power plant because the consulting engineer's "heat balance" figures show that it is economical to do so, we say that it is "modernizing."

American Tobacco Company's Plant

An excellent example of this modernizing idea is in the new power plant of the American Tobacco Company at Durham, N. C. Here we have a plant of the most modern type for industrials, in which the well advertised "toasting" process of Lucky Strike cigarettes plays an important part in the heat balance which made its modernization an economical proposition. To dispel the old idea of exhaust steam as something which you could stick your finger in and get it warmed but wet, picture toasting going on continuously as a by-product of producing power. Here in this plant, steam is being extracted from the turbines at a pressure and temperature well above the required toasting temperature, so that the exact final temperatures at the toasting machines may be controlled to a very accurate degree, which is essential to get just the final result needed in the product.

We present herein pictures and description of the piping system from the boilers to the various apparatus throughout this new and modern power plant.

The plant consists of three water tube boilers of 500 hp. each which furnish steam for two 1,500 kw. turbine generators and steam for heating and process. Boilers operate at 200 per cent above rated capacity. Steam is delivered to the two turbines at 425 lb. working steam pressure and 175 F superheat. Extraction steam is taken from the two turbines at 90 lb. working steam pressure and 100 F superheat. A desuperheater is provided in connection with the 90 lb. extraction steam so as to control more accurately the temperature of steam for process purposes.

Piping in this power plant conveys steam at all pressures from low pressure exhaust up to 425 lb. with 175 F superheat, and these lines vary in size from ⅛ in. to 18 in. O.D.

Figure 1

Fig. 1 is the 12-in. main high pressure steam header. This header is of all-welded construction, made in two sections for convenience of installation. The header is autogenously welded, the details of nozzle construction being as shown in Fig. 2. The header was built in a regular pipe fabricating shop by trained and tested welders and after completion was annealed all over in a furnace to relieve internal stresses and then tested to a hydraulic pressure of 1,200 lb., the piping adjacent to the welds being hammered circumferentially while holding this pressure.

It will be noted that at the center of the header assembly a solid support is rigidly anchored. The ends of the header assembly are, however, supported on adjustable rolls, these being of a construction to allow not only for longitudinal movement but are also provided with means of vertical adjustment after installation (see detail *A*) whereby each support is adjusted to carry its proportion of the load, thereby eliminating strains in the pipe joints.

Nos. 2 indicate the 6-in. leads from boilers carrying 425 lb. pressure superheated steam to the header. (See Fig. 1.)

Nos. 3 indicate the 6-in. leads to the 1,500 kw. turbine generators.

Nos. 4 indicate the 2½-in. and 3-in. steam supply lines from main high pressure header to pressure reducing

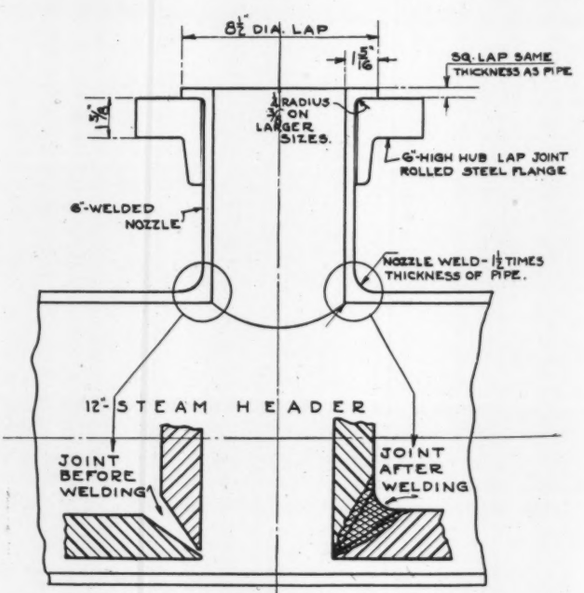


FIG. 2—DETAIL OF NOZZLE CONSTRUCTION IN THE 12-INCH HEADER

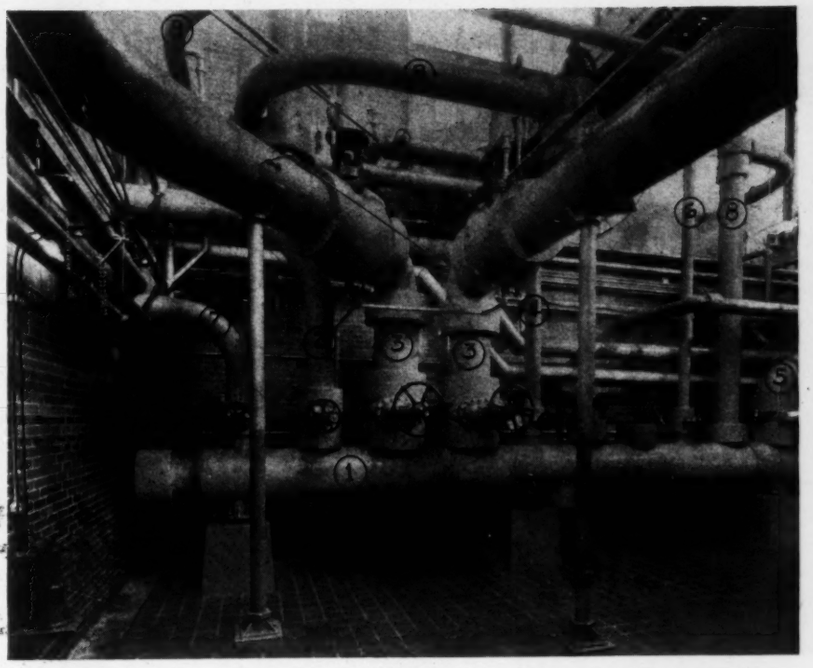


FIG. 3—THE 16-INCH HEADER DISTRIBUTING NINETY-POUND STEAM EXTRACTED FROM THE TURBINES FOR HEATING AND PROCESSING

valves where pressure is reduced to 90 lb., it being then used as an auxiliary supply to the 90 lb. steam header.

Nos. 5 and 6 are steam supply lines to various pumps.

For the high pressure steam lines, extra-heavy open hearth steel pipe is used equipped with high hub forged steel flanges and lap joints. These lap joints are machine-made, being upset in thickness in the process of rolling, are annealed after rolling and are then machined back and front so as to leave them not less than 100 per cent of pipe wall thickness after machining. The faces are "phonograph" finish. Fig. 2, showing the welded-header construction, also shows the type and general dimensions of lap joints used throughout.

Figure 3

No. 1 (in Fig. 3) is the 16-in. welded header into which the 90 lb. steam extracted from the turbines is accumulated for distribution for heating and processing or toasting. The construction of this header is similar to that of the high pressure header excepting that it is of lighter weight materials and the test requirements correspondingly lower. It will be also noted that the anchorage and supporting are the same.

No. 2 (in Fig. 3) is the 6-in. auxiliary connection from main high pressure steam line.

Nos. 3 are the 14-in. extraction lines from turbines.

No. 5 is the 10-in. line for distribution of heating and process steam for toasting. Beyond the point shown in this line is a desuperheater with various connections and instruments so that the required temperatures for various processes are carefully maintained.

Nos. 4, 6 and 7 are 3-in. and 4-in. connections to ash conveyors and stokers, fire pumps and whistle.

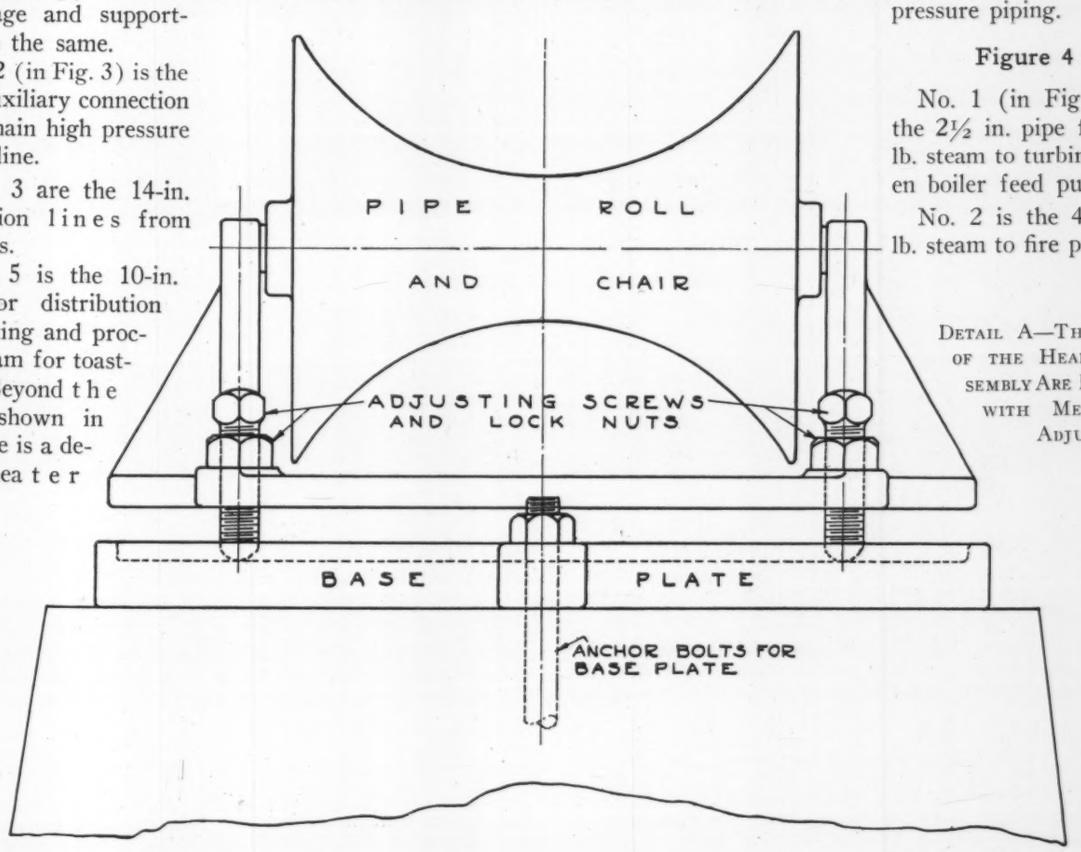
No. 8 is the 8-in. relief connection through relief valve set at 105 lb.

All piping in these lines is standard weight open hearth steel equipped with standard forged steel flanges and lap joints similar to those used in connection with the high pressure piping.

Figure 4

No. 1 (in Fig. 4) is the 2½ in. pipe for 400 lb. steam to turbine driven boiler feed pump.

No. 2 is the 4 in.-90 lb. steam to fire pump.



DETAIL A—THE ENDS OF THE HEADER ASSEMBLY ARE PROVIDED WITH MEANS OF ADJUSTMENT

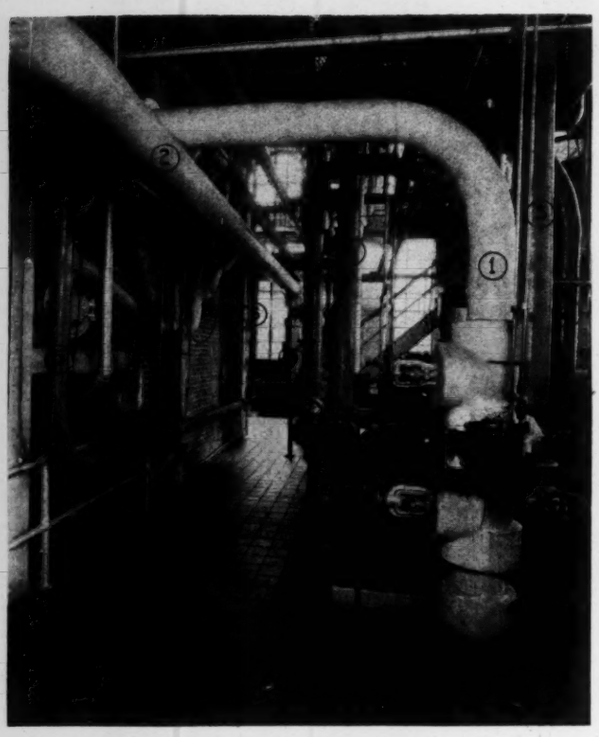


FIG. 4—PIPING FOR THE TURBINE-DRIVEN BOILER FEED PUMP, THE FIRE PUMP, EXHAUST TO FEED WATER HEATER, DISCHARGE FROM PUMPS TO BOILER FEED HEADER, AND SIX-INCH SUCTION TO EACH BOILER FEED PUMP FROM TEN-INCH LINE LEADING TO FEED WATER HEATER

No. 3 is the 6 in.-5 lb. exhaust to feed water heater.

No. 4 is the 4 in. discharge (approximately 450 lb. pressure) from one steam turbine driven pump and one motor driven pump to the 6 in. boiler feed header.

No. 5 is the 2½ in. boiler feed to boiler No. 1 (approximately 450 lb. pressure).

No. 6—one 6 in. suction to each boiler feed pump from a 10 in. line leading to feed water heater.

Piping materials for the boiler feed lines are the same as already described for high pressure steam lines under description of piping in Fig. 1, and for the 90 lb. steam the same as described for piping in Fig. 3. Piping for items 3 and 6 is standard weight with standard cast iron flanges screwed through and refaced.

Figure 5

No. 1 is a 10 in. discharge from each condenser circulating pump to condensers.

No. 2 is a 10 in. discharge from each condenser increasing to 14 in., thence to cooling tower.

No. 3 is a 10 in. atmospheric exhaust from condenser on Unit No. 1.

No. 4 is a 14 in. pipe for extraction steam from turbine No. 2.

No. 5 is a 2½ in. circulating water discharge from oil cooler on turbine to discharge from condensers.

No. 6 is a 2½ in. pipe for cooling water to oil cooler.

Piping in Fig. 5 varies in size from 2½ in. to 14 in.

Items 1, 2 and 3 are designed for pressures up to 125 lb. and is standard steel pipe equipped with standard cast iron screwed flanges on the 10 in. and standard low

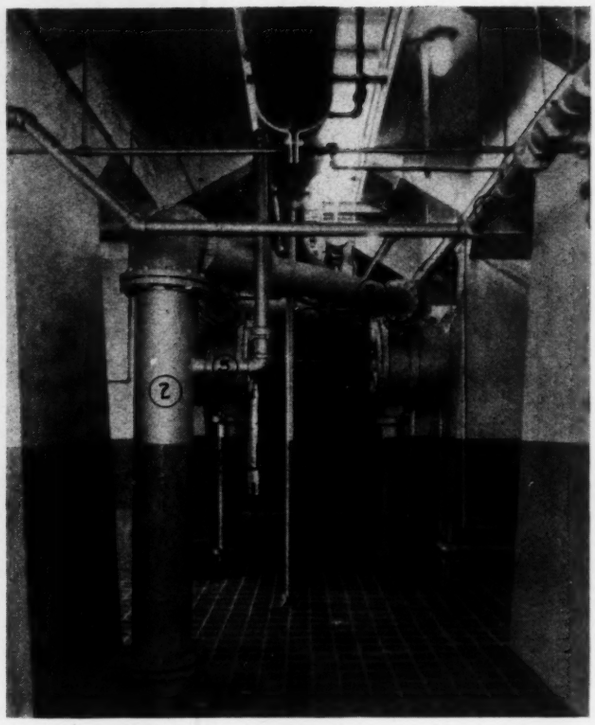


FIG. 5—DISCHARGE AND COOLING WATER PIPING

hub lap joint steel flanges with 100 per cent square laps on the 14 in.

Item 4 is designed for 90 lb. working steam pressure and 100 F superheat (430 F total temp.) full weight open hearth steel pipe ⅜ in. thick equipped with standard low hub lap joint steel flanges with 100 per cent square laps, phonograph finish.

Item 5 is standard steel pipe and standard screwed cast iron fittings.

Item 6 is standard galvanized steel pipe and standard screwed cast iron fittings.

Is it not probable that in many large industrial plants there are unthought of processes where this new kind of exhaust steam can be put to such valuable uses that the heat balance may be decidedly on the profit side?

Prof. Callendar Dies

With the death of Professor Hugh Longbourne Callendar, known all over the world for his investigations relative to the properties of steam and as professor of physics at the Imperial College of Science and Technology at London, the engineering profession suffers a keen loss. Prof. Callendar died suddenly January 22, at the age of sixty-seven.

Professor Callendar started his studies on the properties of steam early in his career. His steam tables, the results of his research work, were accepted almost universally by British engineers and scientists and were adopted by the electrical and allied industries as standard. He took a leading part in the steam table conference, held in England last summer, for the establishment of basic international values for the properties of steam. Callendar was educated at Cambridge. He was elected a Fellow of the Royal Society in 1894.

Air Conditioning a Flour Mill

By Edgar S. Miller

A Modern, Air-Conditioned Flour Mill

THE desirability of atmospheric control has been realized by wheat flour millers for a long time, but it is only within recent years that any serious attempt has been made toward approaching the problem in a scientific manner. A number of European mills are using some sort of conditioning equipment, and in America the subject is provoking more discussion at the present time than any other phase of the milling process.

Many leading mills are now using more or less complete air-conditioning systems, and among the most modern and best equipped is the "Daylight Mill" of the Acme-Evans Company, of Indianapolis, Ind.

The mill building is of reinforced concrete and glass construction, eight stories and basement. The lower floors are used only for packing, storage and loading, the mill proper occupying the fourth to the eighth stories, inclusive. The air-conditioning apparatus is located on the eighth floor.

The equipment consists of an air-conditioning machine capable of full adiabatic saturation, a large, low-speed fan, extended-surface heaters, dirt eliminators, volume dampers, dew-point control, with thermostat-controlled fresh and return-air dampers, and a complete system of distributing ducts capable of conveying the conditioned air to every part of the building devoted to the manufacturing process. Thermostats control both the air volume and the steam to the reheaters. No air can enter the rooms except through the conditioner, a matter of no little significance in connection with the manufacture of flour.

Ton of Air Per Barrel of Flour

It may be a matter of interest to those unfamiliar with the process of flour milling, but who are quite familiar with the ultimate product as it reaches the table in the form of light, palatable bread, that for every barrel of flour milled, something like a ton of air is brought into intimate contact with the particles from which this bread will be made. It is obvious, then, that from the standpoint of sanitation alone, all air used in the milling process should be thoroughly cleansed. The flour particles are of a decidedly porous nature, and their warm, moist surfaces offer an ideal environment for the various species of bacteria and spores with which unwashed air teems. Soot from locomotives and factory chimneys, and filthy dust from the streets, must be effectually removed from the air entering the mill, if the flour produced is to be uncontaminated.

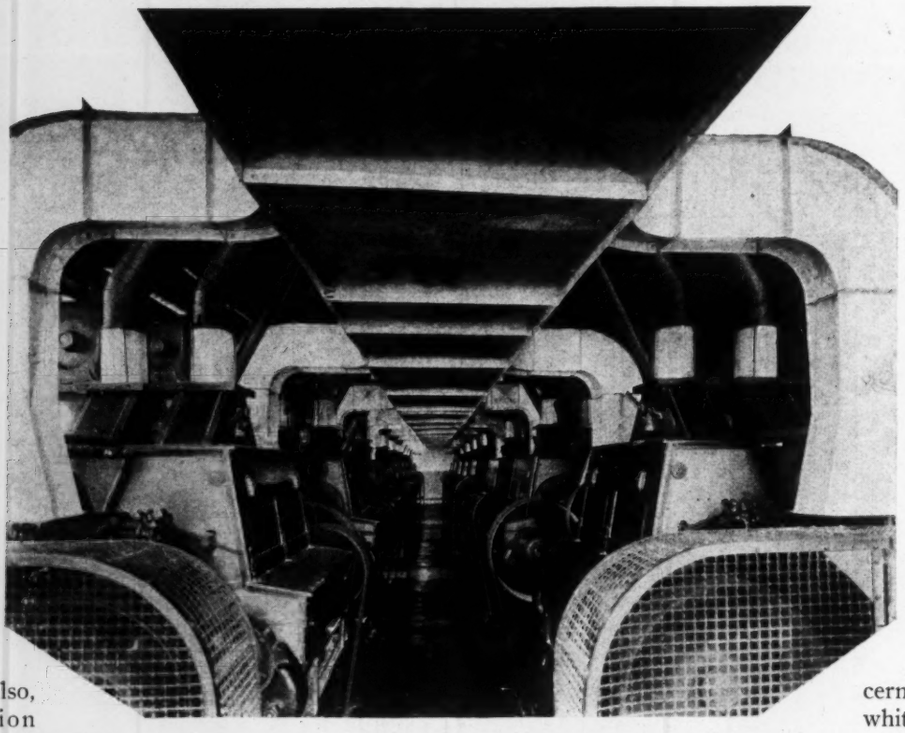
There is, however, still another consideration. This has to do, not with the purity of the air used, important though that phase is, but with the effects of this air upon the products of the mill, and upon the manufacturing process itself. Since, as before mentioned, the products of the wheat berry are porous, contain some free moisture, and therefore present to the air currents comparatively moist surfaces, the air, by virtue of its heat-metamorphic properties, is able to change their very physical nature. This may in no small degree alter the chemical characteristics also, and it may, and often does, exert so powerful an influence upon evaporation as to spell the difference between profit and loss to the miller.

It may be wondered why such enormous quantities of air are used. The principal reason is purification. From the time the wheat is received until the finished flour is packaged, purification, by means of air currents, is almost continuous.

The bran-coat, or covering, of the wheat's true kernel is much like the shell of a nut or the skin of a banana. Since nature provided this covering as a protection against contamination of the true kernel, it is itself always more or less contaminated. A wall designed to keep out dirt is likely to become dirty, and the first thing the miller must do is to scour the dirt from the outside of the grain. This is commonly accomplished by friction, and the dirty scourings are removed by air currents.

Separation of Impurities

To the miller, anything other than the endosperm, i. e., the creamy-white interior of the grain—the wheat's true "kernel"—is "dirt." When the berries are broken up, for the double purpose of reducing the endosperm to a suitable state of fineness and facilitating the removal of the bran, some particles of the latter are inevitably re-



Showing the Air Distribution System on the Roller Floor of the Mill

duced in size also, and separation wholly by means of sieves is impossible. But since the specific weight of the particles is different, a separation can be made by causing air currents to hold the lighter impurities in suspension while the heavier endosperm-meal falls through the meshes of the sieve by gravity.

Again, because wheat is a living organism, extremely sensitive to temperature changes, means for removing some of the mechanical heat generated during the milling process is imperative. Not only do excessive temperatures make sifting difficult; too much heat is certain to stimulate chemical activity in the flour, detracting seriously from its keeping qualities and even impairing its baking value.

Most of the separations of the wheat's products are accomplished by the use of silk cloth. These cloths range as high as 150 meshes to the linear inch, with apertures measuring about .0036 inch across. It is easy to think of these apertures as infinitely small, but modern physics brings out forcibly the fact that size is really only comparative. A flour granule of average size, though perhaps not more than .0025 inch across, actually contains hundreds of starch cells and thousands of colloidal particles. These in turn are made up of an infinite number of molecules, and it is further known that each starch molecule contains thousands of atoms.

Under favorable physical conditions the endosperm, or flour-bearing portion of the wheat, is more easily disintegrated than is the bran-coat or the germ, and from the earliest times milling has depended upon their physical dissimilitude for the separation into the pure and the impure. It might be noted in passing that this classification is not altogether illogical. Wholly aside from its chemical aspects, the bacteria-bearing bran-coat cannot possibly be as immaculate as is virgin endosperm.

Man-kind's demand for white flour is, then, not altogether unreasonable, though it was not originally concerned with the whiteness produced by bleaching. Bran imparts to flour a dull, grayish-brown tint, and for this and many other reasons bran in flour is highly undesirable. Actually, the age-old demand has been for a color which indicates freedom from bran contamination.

A little more than a half century ago the present system of milling first made its appearance. It was then conceived that the best possible separation of endosperm from bran-coat could be made by abandoning the crushing of the wheat berry and attempting to separate the resultant flour. The new method consisted of "breaking" the grain as gently as possible and recovering as separates nearly bran-free particles of the largest possible size. These could be "purified" out of contact with the bulk of the bran. Thus the "patented process" was born, and thus the origination of the term, "patent flour."

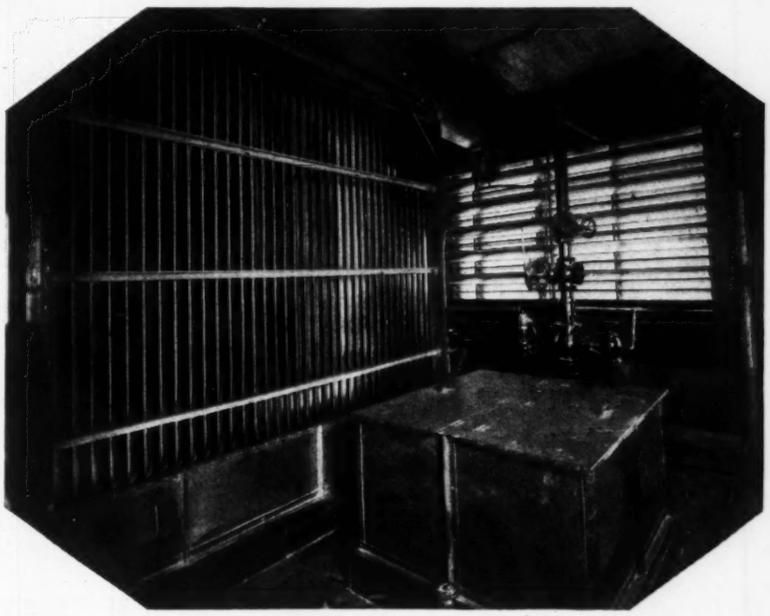
Millers soon learned that the "favorable physical condition" demanded a moisture content of grain in excess of its hygroscopic capacity. Moisture makes all parts of the grain less friable, but the effect upon bran and germ is more pronounced, and the dampening of wheat soon became quite general. Still later it was discovered that the thin cellulose walls which separate the flour cells within the endosperm were also favorably affected by the imbibition of moisture, and the "tempering" of wheat by the employment of both water and time came into being.

But though both the bran-coat and the interior cellulose are toughened by moisture, and though both absorb water with greater facility than do the denser parts, both give up their contained moisture to air more freely. It is obvious, then, that the effect upon the mill stocks of air having a high evaporative power is very pronounced. And since air currents are always tending to undo the very things accomplished by the "tempering," or "con-

ditioning," process—in a larger degree when evaporation is great, and in a lesser when it is small—it should also be obvious that wide fluctuations of temperature and humidity are certain to be fatal to the consummation of the miller's most-sought goal: the maintenance of uniformity of product.

Moreover, because the products of wheat are hygroscopic, and because, at the beginning of the milling process, they always contain moisture in excess of their hygroscopic capacity, the heat metamorphosis coincident to evaporation tends to bring their temperatures down toward the temperature of adiabatic saturation of the air they encounter, or the wet-bulb temperature of this air. Thus, regardless of the original temperature of the grain or the dry-bulb temperature of the air currents, the temperature of the mill stocks must always seek the wet-bulb perature of the air with which they come into contact. If this fluctuates widely, the temperature of the stocks must also fluctuate widely. So also will the threads of the silk-sifting-cloth vary in size, with a consequent variation in area of the apertures.

Fluctuating effective temperatures, therefore, affect the milling process in three ways. The materials themselves are made softer or more brittle with a variation of temperature or with a variation of contained moisture, facilitating or retarding both reduction and separation, and the sifting capacity of the silk cloths is increased or diminished because of a variation in size of the apertures, brought about by variations in the temperature and humidity of the air within the machines.



Part of the Air Conditioning Apparatus

The flooding of the mill with washed air, the wet-bulb temperature of which is automatically controlled, insures air currents with constant evaporative power, as well as a practically constant temperature in the materials being processed. This is assured by accomplishing the major part of the inevitable heat metamorphosis in the air-conditioning apparatus, leaving a definite amount to be done by the air currents in contact with the stocks, and controlling not only the temperature but the ultimate heat content of the air.

The research laboratory of the *American Society of Heating and Ventilating Engineers* has admirably demonstrated that the significant factor in providing a comfortable condition for human beings is the *effective* temperature. This means, not the dry-bulb temperature or the wet-bulb temperature of the air introduced into a room, but the actual heat-equilibrium properties of the stratum of air in intimate contact with the body after this air has been affected both by heat metamorphosis, induced by the movement of a definite quantity of prescribed air over the body, and by isothermal phenomena.

Not only is the human body thus affected; exactly the same laws govern the final effect of thermal phenomena upon any body which is capable of exchanging both water and heat with the air surrounding it.

Before describing the operation of the air conditioning plant in the Acme-Evans Mill, a brief consideration of the effects of weather upon the operation of an uncontrolled mill will be in order.

With an outdoor temperature of 100 degrees (dry-bulb) the mechanically-generated heat liberated within the building is usually sufficient to raise the temperature of the volume of air ordinarily used some 20 degrees. A part of this heat is made latent by the evaporation of moisture from the materials, and this heat is, of course, present in the air in, and leaving, the machines. This air always has a high degree of saturation, and because of its relatively high temperature, conditions which will result in condensation of moisture upon the inner walls of the spouts, elevators, etc., are found. This results in sweating.

The heat not removed by the air currents is dissipated into the rooms, and is sometimes sufficient to raise the room temperatures to 110 degrees, or even higher. With a change in the weather, conditions will change very rapidly, of course, but one characteristic persists: the air within the machines maintains an approximately constant degree of saturation, even though its temperature may vary through a wide range. In other words, the damp materials being processed—the mill stocks—have the function of a humidifier to a limited extent, tending to produce adiabatic saturation in an almost constant degree.

When the outdoor wet-bulb temperature accompanying a 100-degree dry bulb is 75 degrees (a not uncommon occurrence) the air conditioning apparatus in the Acme-Evans mill delivers air with virtually that temperature; namely, 75 degrees dry bulb. The quantity moved is carefully regulated for the purpose of allowing exactly the temperature rise necessary for the relative humidity condition required. Thus, if 70 per cent relative humidity is desired, the mill is supplied with enough conditioned air with a temperature of 75 degrees to insure a rise of 11 degrees, and no more.

Of course, so long as this "spread" between the orig-

inal and final temperatures of the conditioned air is maintained, relative humidity must remain constant. With these temperatures kept constant the wet-bulb temperature in the mill must remain constant, resulting in unvarying temperatures in the mill stocks.

The properties of the conditioned air are fixed by controlling the relative proportions of fresh air and recirculated air, with the concomitant recovery of the necessary amount of mechanically-generated heat. When this becomes insufficient, as it does in very cold weather, the dew-point control automatically admits steam to a preheater.

It is possible to maintain an apparatus temperature of about 75 degrees, with an observed room temperature of 86, with 70 per cent relative humidity, in any weather. It is usual, however, to lower these somewhat in winter, since no useful purpose would be served by expending fuel to maintain higher wet-bulb temperatures than are necessary.

Fluctuations are of all things the most to be avoided, and some millers favor the maintenance of two constant but different conditions at different periods of the year. It is true, perhaps, that the best possible wet-bulb temperature obtainable in summer by the employment of evaporative cooling only, is somewhat too high. Certainly this is true unless the milling system is arranged to meet the condition, and under certain circumstances the installation of apparatus for dehumidification in the summer is justified.

Since the air leaving the conditioner is always saturated, changes in the relative humidity out-of-doors have no effect upon the mill. If the air coming to the machine is already nearly saturated, the amount of water vaporized is small, while during the afternoon of a hot, dry day the quantity is quite large. The significant point is that evaporation from the materials being processed is always the same, the humidifier taking care of the fluctuating evaporative factor. In other words, stocks still function as does a humidifier, but only in a slight and very constant degree. Excessive evaporation from them is prevented by causing the humidifying apparatus to perform the greater part of that work, and water from the city mains, or from wells, is substituted for the moisture for which the miller paid some two cents a pound, as wheat.

Winter operations are as simple and of fully as much moment. When air at zero, or below, is heated to 80 or 85 degrees in the mill, its dew-point temperature would necessarily remain in the neighborhood of zero except for the addition of water vapor, either from the mill stocks or from the water sprayed in a conditioner. The conditioner adds this water at a price infinitesimal when compared with the price of wheat. Of chief importance, however, from the standpoint of quality and uniformity of product, is the maintenance of virtually constant effective temperatures within the milling process. Uncontrolled, effective temperatures may vary 15 or 20 degrees within a period of a few hours, and the consequent disruption of all the mill adjustments is a serious matter. True, the operatives can and do make readjustments, but necessarily only after some damage has been done.

Removal of Corrosive Fumes

NOT only is the effect of corrosive fumes on operators a reason for their removal from pickling rooms, bright dipping plants and similar locations, but also is the fact that these fumes have a severe effect on the materials of construction in the building.

Those processes which involve nitric acid, such as bright dipping, silver stripping and the like, offer possibly the most difficult problems.

While aluminum and the higher chrome steels satisfactorily resist pure nitric acid, it is frequently the case that the presence of mixtures of other acids or salts will cause severe attack.

Earthenware equipment is frequently satisfactory for resisting corrosive fumes but the inherent difficulties, due to its low strength, often offer possibilities of breakage when employed in exhaust fans.

Nearly all operations from which corrosive fumes emanate involve more than one corrosive agent, thus complicating the selection of suitable material. In many cases a ferro silicon alloy is used.

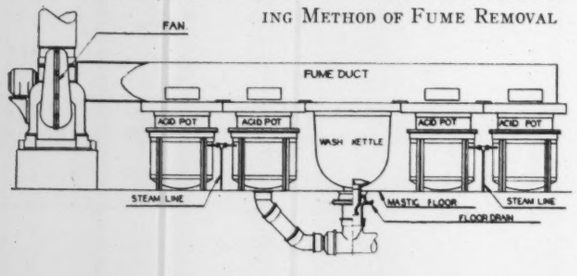
In such operations as bright dipping and silver stripping, it is necessary that a high velocity of air be maintained across the surface of the acid vessel, this velocity being sufficient to carry all of the fumes directly into the exhaust duct.

The breathing of nitric fumes has a very injurious effect on the operator.

The accompanying cut shows a typical installation for silver stripping. It consists of five pots, four of which contain nitric acid which is kept hot by steam in the steel jackets. The fifth, or center, pot is merely a wash vessel which, however, must be of an acid resisting material because the wash water soon has an appreciable acid content. The fan and drain are made of a ferro-silicon alloy.

It will be noted that the duct for the fume exhaust runs close behind the vessels. Opposite each pot is a slot which is as close as practicable to the surface of the acid. The exhaust fan is of sufficient size to create adequate velocity so that fumes are drawn quickly from the entire surface of each pot and taken into the duct. A similar method should be used whenever it is desired to remove the more severe acid fumes. This plan also eliminates interference with operations, which is frequently caused by overhead hoods.

Bright Dipping, Plating or Stripping Installation Showing Method of Fume Removal



The Flow of Superheated Ammonia

in One Inch, Extra Heavy Black Steel Pipe

By John C. Reed and Edgar E. Ambrosius

Introduction

THE following data were obtained as a portion of an investigation on the flow of superheated ammonia undertaken by the authors in the Department of Mechanical Engineering at the University of Illinois.

The authors are continuing the work and the final results will be published in a bulletin of the Engineering Experiment Station of the University of Illinois. A complete report of the investigation up to Oct. 1, 1929 was presented to The American Society of Refrigerating Engineers at their December meeting and appears in the February issue of their journal.

In refrigeration, where superheated ammonia is to be used as a working medium, a knowledge of the magnitude of frictional losses is necessary for the most economical choice of pipe sizes and equipment. The ideal pipe size to use is that one which strikes an economic equilibrium between cost of material, cost of installation, and the cost of power to overcome the frictional resistance.

The determination of the frictional resistance embodies the use of the friction factor, the value of which may be determined for any given set of conditions. The friction factor is directly dependent upon the pressure drop. If a large pressure drop occurs in the suction line of an ammonia compressor, there is a loss of refrigeration. A pressure drop in the discharge line means an increase in horsepower required. In order to overcome these losses, there must be supplied a larger cylinder displacement and also additional horsepower must be furnished. The increasing of either or both of these factors means an increase in the cost per ton of refrigeration. Economically, it is desirable to reduce these factors to a minimum, thereby reducing the cost per ton of refrigeration to a minimum.

Object

The object of this investigation was to determine the relation between the friction factor and Reynolds' number when superheated ammonia is circulated in one inch, extra heavy, black, steel pipe under conditions of pressure and temperature similar to those encountered in refrigerating practice.

A study of the problem will show that friction is the chief factor in producing the pressure drop. Not only is there friction between the fluid and the surface of the pipe, but also within the fluid itself. This internal friction manifests itself when turbulent motion exists within the pipe, and is dependent upon the viscosity of the fluid flowing.

In just what proportions these two types of friction are present is not known, so no attempt will be made to arrive at a separate friction factor for each at this time. The results of experiments do show, and it is also verified in this work, that the frictional resistance varies approximately with the density of the fluid, the square of the velocity, and the pipe diameter. Since the coefficient of friction is not a constant for all conditions of flow, it may be inferred that the resistance varies as some power other than unity with the density, and as some power other than two with the velocity. Pipe diameter and roughness also play an important part in producing pressure drop. In general, the roughness effect diminishes with an increase in pipe diameter.

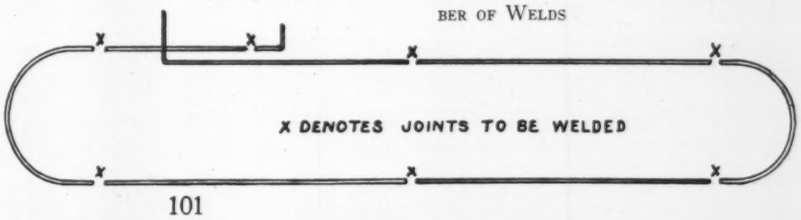
FIG. 1—END VIEW OF PIPE JOINT TO BE WELDED SHOWING HOW THE AVERAGE DIAMETER WAS MEASURED

A definite relation occurs between the friction factor and a ratio known as Reynolds' number. This ratio is composed of the dimensions of the pipe, the average velocity of the fluid, the density and absolute viscosity of the fluid.

Maxwell defines viscosity as follows: The viscosity of a substance is measured by the tangential force on a unit area of either of two horizontal planes at a unit distance apart, required to move one plane with unit velocity in reference to the other plane, the space between being filled with the viscous substance. The viscosities of gases, unlike those of liquids, increase with temperature, but are practically independent of the pressure. Also, viscosity is a fundamental property independent of the particular method used in its measurement. With this knowledge in mind, all the values of absolute viscosity that could be found were plotted against temperature. The values of viscosity required in this work were taken from this curve corresponding to the average temperature in the test section.

Having obtained the necessary data, they were analyzed according to the dimensionless homogeneity theory; that is, analyzed according to the friction factor and

FIG. 2—PLAN OF TEST SECTION DENOTING POSITION AND NUMBER OF WELDS



Reynolds' number criterion, since they are both dimensionless quantities.

It is an established fact, taken from the work of Reynolds' and others, that stream line motion exists, when the value of Reynolds' number $\left(R = \frac{D\,U\,s}{N}\right)$* is less than 2500. Turbulent motion takes place for values of R above 2500. Under certain conditions, we may have either turbulent or stream line motion for values of R between 2000 and 3000.

By using the proper values of the friction factor $\left(f = \frac{2\,h\,D\,g}{4\,L\,U^2}\right)$* the pressure drop can be calculated.

The value of f, is a function of the velocity, pipe diameter, length and condition of pipe. Fortunately, however, f is not an independent function of these variables, but of the ratio, $\frac{D\,U\,s}{N}$. Another fortunate feature is that this ratio has no dimensions. It is, therefore, a function of variables combined in such a way to give a dimensionless result.

It is upon this basis that the results of this investigation were analyzed and the value of the results to one interested in the selection of pipe sizes are shown in the conclusions of this article.

The superheated ammonia gas for these tests was supplied by a twin vertical single-acting ammonia compressor in the mechanical engineering laboratory. The test section, consisting of 100.85 ft. of one-inch, extra-heavy, black, steel pipe, was located between the compressor and condenser. This size of pipe was selected owing to the limitations determined by the capacity of the compressor. With larger pipe, the capacity was not sufficient to obtain velocities necessary to give a fairly extensive range of Reynolds' numbers. Smaller pipe was not considered comparable with the sizes used in commercial practice. In addition, the location selected insured that the ammonia could be supplied with the required range of superheat and pressure.

The pipe was commercial pipe purchased in the open market. Before being installed the diameter was measured at both ends of each section and was carefully inspected. The ends were milled square and all burrs removed, and the sections were then butt welded in place as shown in Fig. 2. Subsequent examination proved that this procedure gave joints free from internal burrs and other irregularities. The pipe may therefore be regarded as "smooth pipe." The average diameter as determined by the mean of the 89 separate measurements was 0.955 in.

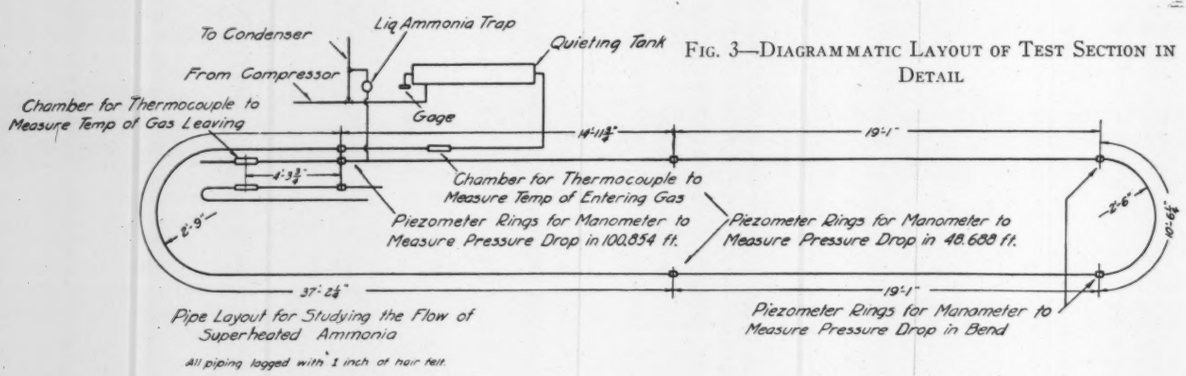

To Condenser
Liq Ammonia Trap
Quieting Tank
From Compressor
Chamber for Thermocouple to Measure Temp of Gas Leaving
Gage
14'-11⅞"
19'-1"
4'-3⅞"
Chamber for Thermocouple to Measure Temp of Entering Gas
2'-9"
Piezometer Rings for Manometer to Measure Pressure Drop in 100.854 ft.
Piezometer Rings for Manometer to Measure Pressure Drop in 48.688 ft.
2'-6"
10'-6¼"
37'-2⅛"
19'-1"
Pipe Layout for Studying the Flow of Superheated Ammonia
Piezometer Rings for Manometer to Measure Pressure Drop in Bend
All piping lagged with 1 inch of hair felt

FIG. 3—DIAGRAMMATIC LAYOUT OF TEST SECTION IN DETAIL

The long radius bends used were carefully made and inspected to insure that the cross-section remained uniform. After installation, with all gages and manometers in place, the line was subjected to 200 lb. per sq. in. air pressure and tested for leaks by means of soapy water. Before being placed in service ammonia was admitted and the line again tested for leaks by means of burning sulphur wicks, and then covered with approximately one inch of hair-felt.

The Tested Section

The exact dimensions and layout of this section will be found in Fig. 3. The test section could be by-passed—but under testing conditions, the gas came from the compressor into a receiver or surge tank inserted just before the test section. This tank was ten inches in diameter and six feet long. These dimensions were ample, since pulsations in the test section were practically eliminated.

The ammonia gas discharged from the receiver into a small mixing chamber (shown in Fig. 3 and detailed in Fig. 4) containing a thermocouple for determining the temperature of the gas entering the test section. All temperatures were meas-

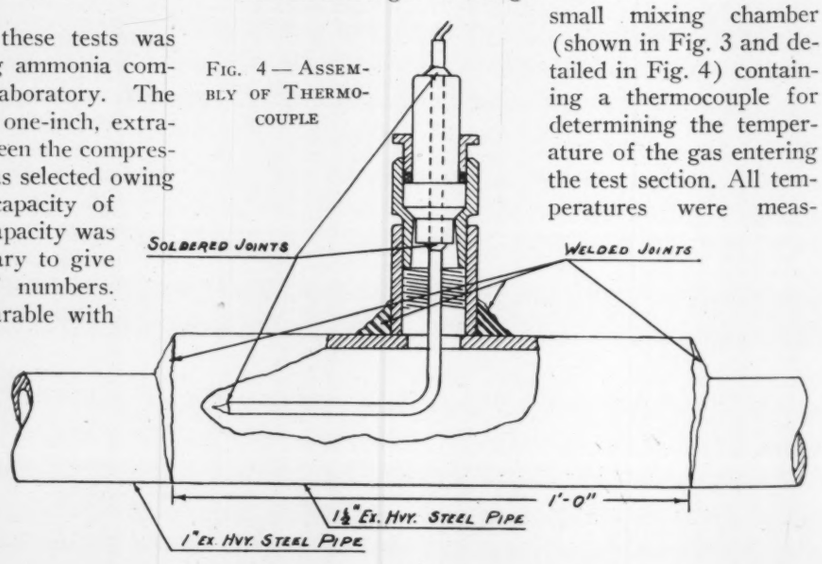


FIG. 4 — ASSEMBLY OF THERMOCOUPLE

* See nomenclature at end of article.

ured by means of calibrated copper-constantan thermocouples made of No. 34 B. S. gage wire. A single cold junction at 32 F was used, this being a method suggested by Foote, Fairchild and Harrison in "Pyrometric Practice," *U. S. Bureau of Standards*, Paper No. 170. The object in having the thermocouple located in a chamber larger than the normal pipe size was to provide ample space for thorough mixing of the gas. With this arrangement, it was thought that there could be no question as to the temperature of the gas entering the test section. The temperature of the gas leaving was obtained in a similar manner.

The manometer pressure readings were taken from piezometer rings which were welded integral with the piping, as shown in Fig. 5. These rings consisted of a circumferential passage which communicated with the interior of the pipe by four No. 30 drilled holes, these occurring at equal intervals around the pipe. This arrangement insured that a correct average pressure would be obtained and the blocking up of any single hole would not cause failure in obtaining correct manometer readings.

U-tube manometers were used to measure the pressure losses. These manometers were built of steel to withstand the high pressure existing in the system. The details of their construction may be observed in Fig. 6, and with one exception, the front and back were enclosed with heavy plate glass. The scales were glued to the front side of the rear glass. All liquid ammonia that condensed in the manometers before a test was boiled off before any readings were taken. Three such manometers were used and their location may be noted in Fig. 3 as well as in Fig. 7.

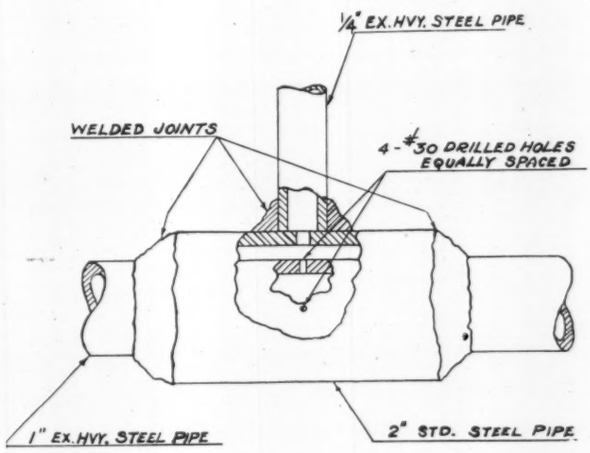


Fig. 5—Assembly of Piezometer Ring and Manometer Connection

A Bourdon type ammonia pressure gage was used to determine the static pressure in the test section. This gage was located centrally so that it recorded the average pressure in the test section.

The gas after passing through the test section was discharged into a trap which was inserted at this point to collect any liquid that might form. The trap served as a very positive check on the fact that superheated ammonia was flowing at all times for never was there any liquid in this trap.

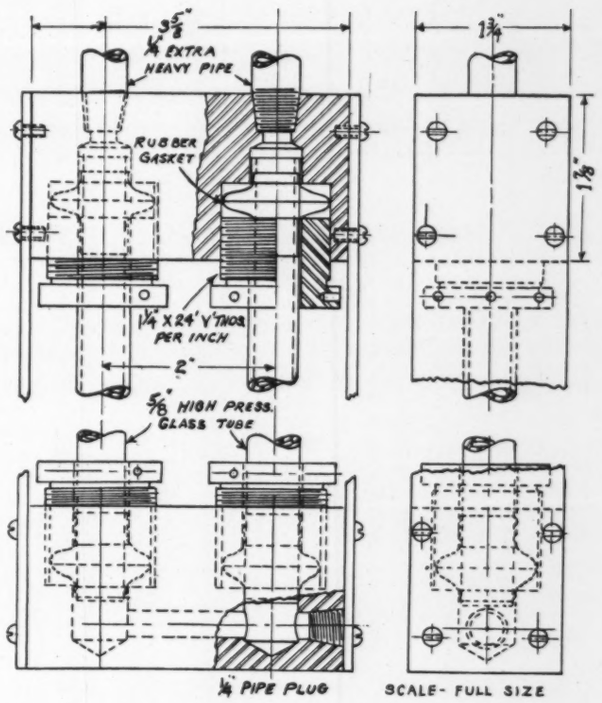


Fig. 6—Mercury Manometer Constructed of Steel to Withstand High Pressures

Diagramatic Layout of Entire System

Fig. 7 illustrates diagramatically the location of all equipment used in this investigation. By starting with the arrow, leaving the compressor and following each succeeding arrow, the ammonia may be traced through each piece of apparatus and thus the entire cycle of operations may be observed.

The compressor used is shown in Fig. 8. It was engine-driven and had a speed variation from 80 to 100 revolutions per minute. The gas discharged from the compressor into an oil trap which worked quite successfully and only a few times was there evidence of oil in the manometers.

The condenser[1] used was of the horizontal, multitube, multipass type. It consisted of two shells, each containing seven tubes about 12 ft. long and about two inches outside diameter. A motor-driven centrifugal pump was used to furnish cooling water for this condenser.

The amount of ammonia condensed in a given time was obtained by direct weighing,[2] in one of two drums located on platform scales as indicated in Fig. 7. The piping from each of these drums was suspended horizontally and was connected to the supply valves by means of long sweep bends. This arrangement gave enough flexibility to the weighing system so that the scales were sensitive to a very small change in weight on the scales. A suitable system of valves permitted either drum to be filled with ammonia while the other was being emptied. The ammonia from the latter went directly to the expansion valve. Obviously the plant operation was continuous, while the weighings were intermittent.

From the expansion valve, the ammonia circulated

[1] "Heat Transfer in a Multitube-Multipass Condenser." Presented at the December meeting of the A. S. R. E. in 1928.

[2] See Bulletin No. 171 Engineering Experiment Station, University of Illinois.

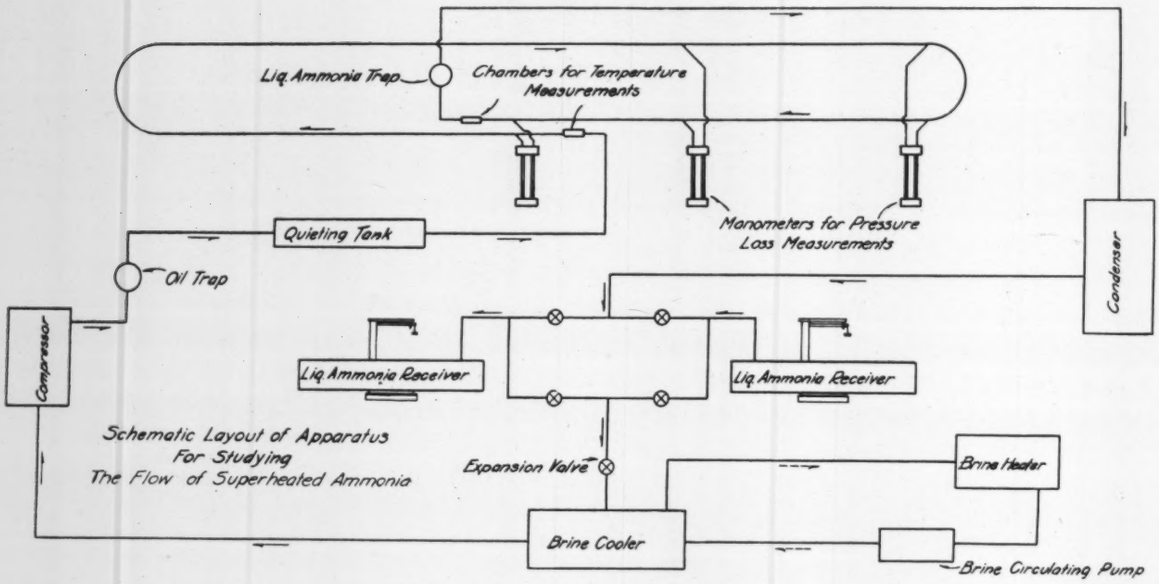


FIG. 7—SCHEMATIC LAYOUT OF ENTIRE PLANT

through the brine cooler. The brine was circulated by means of a centrifugal pump, the amount being controlled by a throttle valve at the pump outlet.

From the cooler the ammonia returned to the compressor to begin another cycle.

Test Procedure

The plan pursued was as follows: The plant was put in operation and run several hours with someone in constant attendance to bring the brine temperature and suction pressure to the desired values. After these conditions were obtained, the plant was operated until equilibrium was maintained in the test section. That is, tests were not made until the pressure and temperature became constant and remained so for at least an hour.

A test, or run, was the time required for the collection of 50 lb. of liquid ammonia leaving the condenser. For very slow rates of condensation the time for the collection of 25 lb. constituted a run. The pressure and pressure drop in the line were recorded. Having the weight and pressure, the volume and velocity were calculated. Room temperature as well as several other temperatures were recorded at intervals during the test, while the barometer was read only once. Care was exercised that no liquid ammonia was present. As many temperature and pressure readings were secured as convenience would permit and the number of readings taken ranged from five to twelve, which were averaged for values used in calculations.

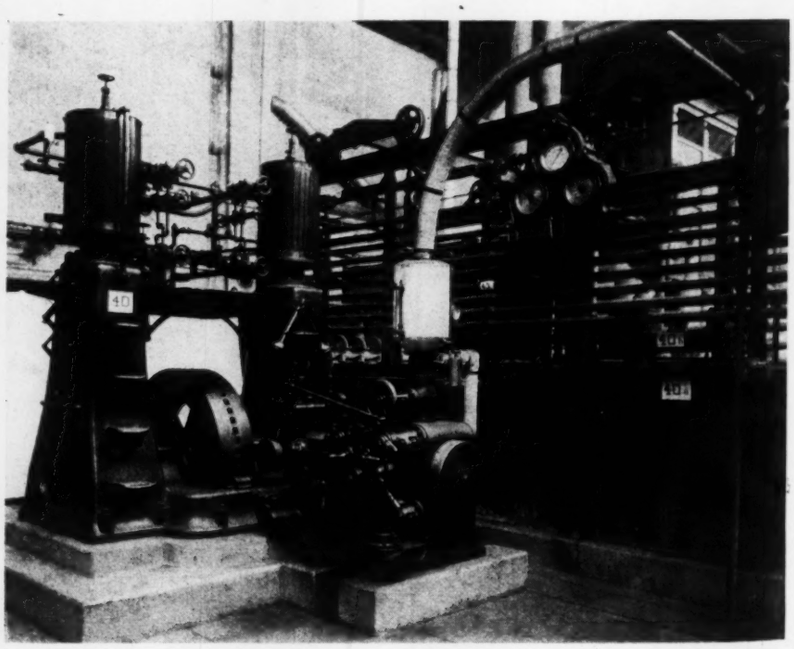


FIG. 8—THE COMPRESSOR USED

Several factors entered into controlling the velocity of the gas, and consequently several means of varying it. The best control of the velocity was by means of the expansion valve. Also if the suction pressure was varied, the amount of ammonia passing through the system would likewise vary. Then, if the amount of water flowing through the condenser be regulated, the pressure could be controlled. That is, an increase in cooling water decreased the pressure in the condenser, and hence, decreased the discharge pressure at the compressor. As a decrease in pressure is accompanied by an increase in volume, the velocity was regulated by this method.

The suction pressure was maintained constant by keeping the brine temperature constant. This was accomplished by heating the brine with low pressure steam.

Conclusions

1. The results show that the pressure drop in the test section varied almost as the square of the velocity.
2. An important relation, however, was that between

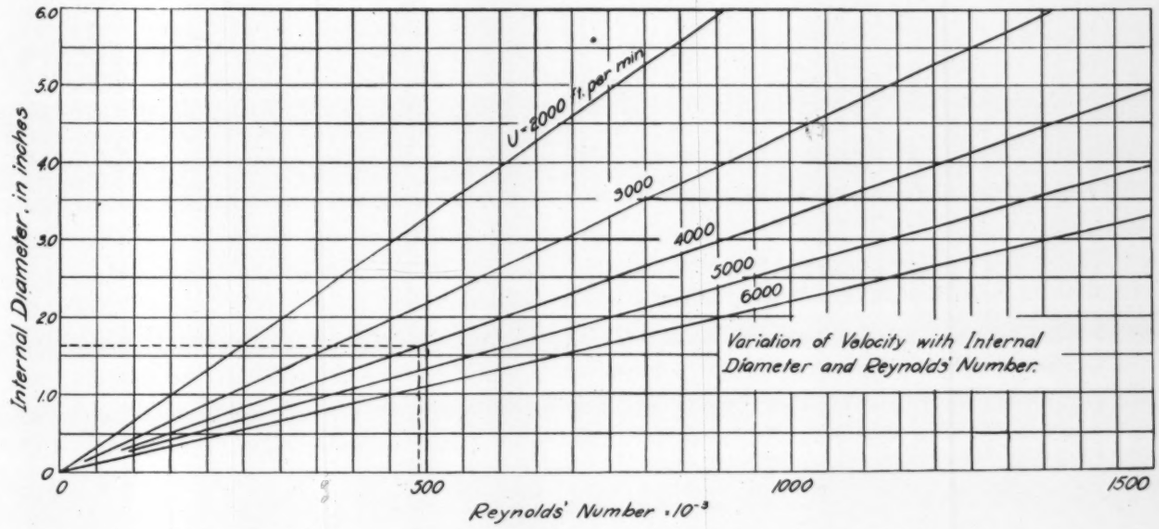


FIG. 9—VARIATION OF VELOCITY WITH INTERNAL DIAMETER AND REYNOLD'S NUMBER

the friction factor and Reynolds' number. A plot of the authors' experimental values with those of other experimenters, indicate a close check. This leads to the conclusion that the general theory of flow which involves the expression of friction factor and Reynolds' number is applicable to superheated ammonia flowing in circular pipes.

3. The curve proposed by McAdams for air, steam, water, and oil in steel and cast iron pipe, is applicable in the case of superheated ammonia within the range of this investigation.

4. It may be further concluded that the results of the tests help to substantiate the theory that a common relation exists between friction factor and Reynolds' number for the flow of all fluids.

5. The direct application of this investigation may be illustrated by the solution of a typical problem.

Given: 152 ft. of 1½ in. (internal diameter 1.611 in.) ammonia discharge pipe line and a velocity of 4000 ft. per min.

Required: To find the pressure drop.

A graph shown in Fig. 9 illustrates the variation of velocity, with internal pipe diameter as the ordinate and Reynolds' number as the abscissa. Reynolds' number is 490,000 at the intersection of the inside diameter of 1½ in. pipe, and a velocity of 4000 ft. per min.

A tentative curve for friction factor, f, against Reynolds' number, R, is shown in Fig. 10. It is the author's belief that as more data are collected this curve may need slight revision. A Reynolds' number of 490,000, Fig. 10, represents a friction factor of 0.00505. Having the friction factor, the pressure drop may be calculated from the formula,

$$p' = 0.00247 \frac{f\ L\ U^2}{D'}$$

$$p' = 0.00247 \times 0.00505 \times 152 \times \frac{\left(\frac{4000}{60}\right)^2}{1.611}$$

$p' = 5.22$ lb. per sq. in.

This demonstrates a method of obtaining the drop in pressure in a given pipe line. Having determined this drop, one is better enabled to select the most economical pipe size. The same procedure may be followed in determining the drop in pressure in the suction line also.

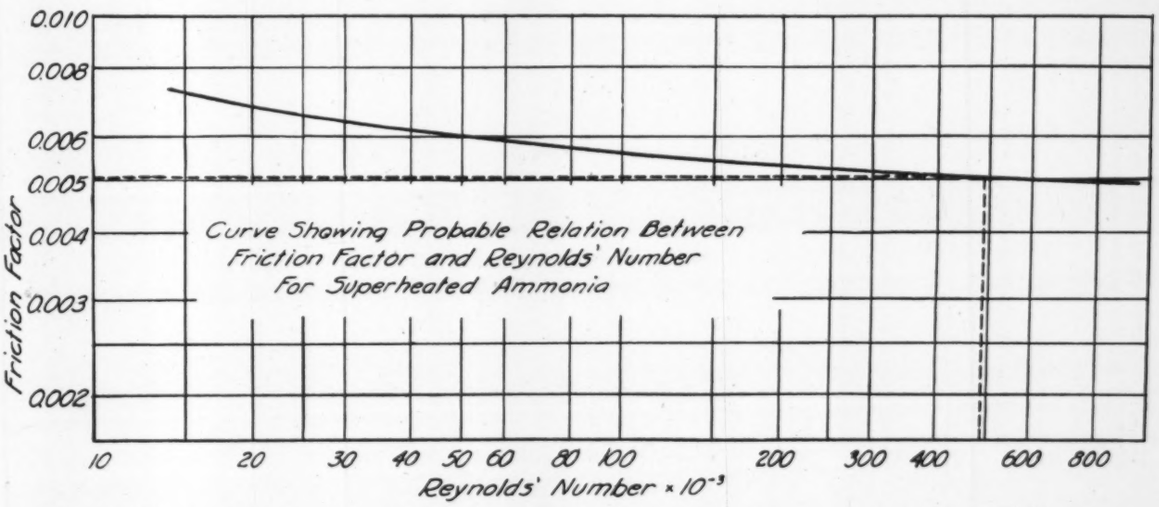


FIG. 10—PROBABLE RELATION BETWEEN FRICTION FACTOR AND REYNOLD'S NUMBER FOR SUPERHEATED AMMONIA

Nomenclature

p'=drop in pressure in lb. per sq. in.
f=friction factor.
L=length of pipe in ft.
U=velocity of gas in ft. per sec.
D'=internal diameter of pipe in in.

$$f = \frac{2hDg}{4LU^2}$$

h=loss of head in ft. of gas flowing.
g=acceleration of gravity in ft. per sec.^{2}
D=internal diameter of pipe in ft.

$$R = \frac{DUs}{N}$$

R=Reynolds' Number.
s=density in lb. per cu. ft.
N=absolute viscosity in lb. per ft. sec.

Electric Air Heating

By H. L. Stiles

THERE is a general impression held by the public that electric air heating is a convenience which for the present is far too costly as compared with combustion fuels. This is a natural impression and is one which is held even by many associated with the electrical industry in spite of the fact that electricity is rapidly being adapted for auxiliary and main heating here and abroad.

Failure to make thorough analysis of the problem is the principal reason for the prevalence of this impression. There is no disputing the fact that ordinary power rates established for lighting and the use of small appliances are not compatible with electric air heating. In fact, no extensive amount of heating should be done on rates which exceed 2½ or 3 cents per kw.-hr. This is a rather broad statement which requires qualification. For instance, one of the main factors which limit the price which can be paid for air heating electricity, is the type of service to be supplied. If the electric air heaters are to be used for but a few hours during the early fall or late spring, for occasional or auxiliary heating, a higher rate could be paid than for main or continuous service. It must also be considered that the flexibility of electric heaters makes it unnecessary to heat any part of the building except spaces in use.

A Number of Installations

Although electric heating does cost more than combustion fuel heating, the cost is not out of proportion where the proper rates have been established. That this is true is proved by the increasing number of installations.

The Benicia High School installation at Benicia, Calif. with 216 kw., the Clarksburg High School of Clarksburg with 386 kw., and the Tamalpais High School in Mariu County with 450 kw. are a few examples of public and private school installations in California. At Turlock, California, there is a nurses' home and a Woolworth store with heating loads of 46 and 27 kw. each. San Luis Obispo and Buelton are two all-electric towns wherein practically all domestic, public and industrial heating and cooking are done electrically. In the Vernon manufacturing district near Los Angeles, there are a large number of office installations.

Radiant and Convection Heaters

There are two general types of electric heaters on the market. Almost everyone is familiar with the radiant heater, sometimes referred to as a "spot" heater. This type of heater is ideally adapted for short periods of operation, where immediate heat is desired, but should not be used for raising and maintaining the temperature of the air. For "maintained-temperature" heating, as this is called, the type commonly known as convection heater has been developed. It differs from the radiant heater in that its units operate at a black or "non-glowing" temperature. Convection heaters are designed to permit a large volume of air to circulate over the units and be heated. The air continues to recirculate through the heater until the room temperature is raised to any desired degree. The heater may then be turned to a lower heat, manually, or by an automatic thermostat which has been set to control some predeter-

A View of San Diego

mined temperature. The percentage of radiation from this type of heater need and should not exceed that of a steam radiator.

The convection and radiant types of heater have been used very extensively through California. The minimum temperature in that state very rarely falls below freezing and remains there for but a short time. As the kw.-hr. rate averages between 1½ and 2 cents, electric heat may be used exclusively for homes, offices and public buildings.

The convection heater, because of the absence of radiant heat, is being more and more used for stores, public buildings, libraries and schools.

Central Stations Studying Possibilities

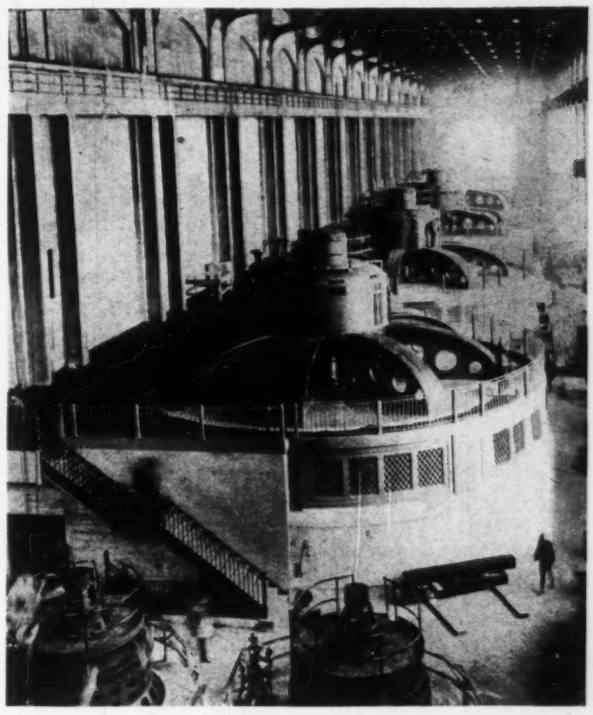
It has been often said that electric air heating will be restricted to the Pacific Coast states where there is an abundance of hydro-electric power. It is generally believed that water power provides cheaper current, but this is not always true. In some instances steam power plants of modern design are capable of delivering current to the customer at a lower rate than is possible with hydro. This fact has been lately demonstrated by one of the largest power companies in California which is actively engaged in building up its air heating load.

There are already a number of power companies in the northern states that are investigating the practicability of electric heat. It is expected that rates attractive for electric heating will be offered to the public to enable them to take advantage of this type of service, at least for auxiliary use. The chief objection by the central stations to this use of current is that the load is seasonal; that is, during the cold periods the load is heavy and pyramided on the industrial and other domestic loads. During the warmer seasons the load is absent with no substitution to take its place. This is a genuine deficiency, particularly in the localities subject to severely cold weather and where, consequently, the power load would be unusually heavy.

To circumvent this difficulty there has been formed a company to exploit the possibility of using night storage heaters which would be turned on during the night hours when the other load is low.

The use of "off-peak" power for air heating is not a new idea either here or abroad. Probably one of the first experiments in this country along this line was started in 1914 at Seattle, Washington, and reported in the Journal of Electricity of June 15, 1919. Although the results indicated that night storage heating was a solution to the central stations' problem, no great development was made at that time.

An electrical heating company of Philadelphia has within the last few years installed a number of night storage electric air heating systems in Fort Wayne, Indiana. The heating systems installed consisted of one or several accumulators or large tanks filled with water, well insulated, and equipped with immersion type electric heating units. These units are automatically turned on during the night hours and heat the water to any predetermined temperature, when they are again turned off. As water has a very high specific heat, sufficient heat can be stored during the night for use during the following day. The heat may be distributed by circulating warm air in ducts or by circulating the hot water. Either system has proved satisfactory.



A Large Hydro-Electric Plant

Estimating Heater Sizes

For estimating the required heater sizes for heating, the method standardized by the American Society of Heating & Ventilating Engineers for calculating heat losses is used. The total calculated loss in B.t.u. is directly converted into the estimated kw. capacity by dividing by 3412. No allowance need be made for inefficiency. To provide for rapid preheating, 15 to 20 per cent is added to the amount calculated to maintain the required indoor temperature. This creates a condition comparable to the increased condensation of a radiator due to the lower room temperature at starting. With electric air heating this is necessary in the event that the heating is done intermittently, in which case the starting temperature may be relatively low. This over-capacity will not increase the total kw.-hr. consumption but will prove to be an economy.

A simple and easy means for comparing the relative costs of electric air heating in two separate regions may be made through the aid of the degree-day unit. For instance, San Francisco has 3450 degree-days, while Dallas has 2455. Although Dallas has the greater temperature fluctuation over the year, the total heating is less. This would indicate, assuming the same power rates existed in both localities, that electric heating would be cheaper in Dallas than San Francisco, where electric heating is already in common use.

Trap Installation for Process Work*

By T. H. Rea

A FEW years ago the majority of traps for industrial heating work operated under quite different conditions than traps for process work. With the advent of the unit heater and with the hundreds of new applications for steam in process work, there is no longer a well-defined boundary between heating and process service. In fact, ordinary heating service can well be

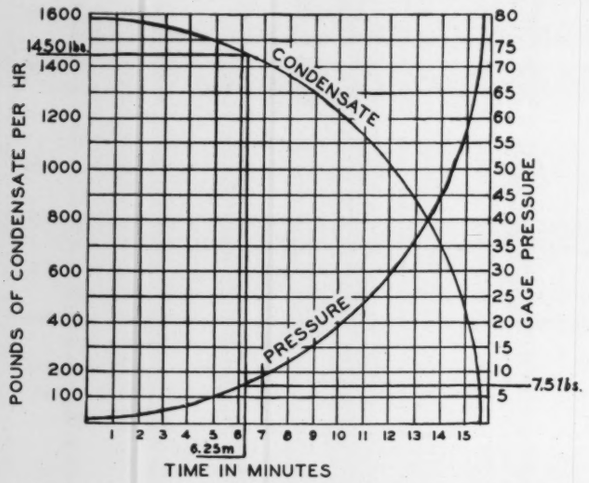


FIG. 3—PRESSURE AND CONDENSATE VARIATION IN BATCH PROCESS WORK

considered as a division of process heating. Process steam heating might be defined as the condensing of steam in suitable pipes or chambers designed for transferring the latent and sensible heat of steam to solids, liquids, or gases. Industrial heating falls within the scope of this definition.

Two Kinds of Process Heating

Process heating work can be classified in two divisions; first, batch or intermittent processes, and second, continuous processes. It is admitted that the typical continuous process application will start and stop and to that extent is intermittent. However, further discussion will reveal that there is a sound basis for this classification.

In batch process work, the material to be heated is retained in the heating unit until the process is completed. Sometimes the material to be processed is introduced into the machine or heating unit after the unit has been brought up to temperature; more frequently the material must be charged into the heating unit before steam can be admitted. Under this second condition, the demand for steam is very heavy at first and the pressure drops sharply. Hence a steam trap must be used that has a large enough orifice to handle the flood of condensate at low pressure differential. As the process advances, the temperature of the material and the unit itself will rise, cutting down the rate of condensation and allowing the pressure to build up. Toward the end of the process, the steam requirement may be small and the pressure in the unit practically equal to the pressure in the steam supply line. Not only must the valve orifice be large enough to handle the peak load of condensate at the low initial pressure but also the operating mechanism of the trap must be powerful enough to pull the discharge valve away from this large orifice at the maximum steam pressure.

High and Low-Pressure Traps on Same Job

When the amount of condensate and the steam pressure vary between wide limits, a high and a low-pressure trap may be used on the same job. The valve orifice in the low-pressure trap is large enough to handle the initial flow but the mechanism will not pull the valve away from the seat after the pressure rises beyond a certain point. By this time, the amount of condensate should be low enough to be handled by the small valve in the high pressure trap. Examples of this extreme service on which a high and low pressure trap can be used to advantage are sulphite pulp cookers and large vulcanizers in the rubber industry.

Continuous Process Work

On the other hand, the load on a continuous process trap is comparatively uniform.

The dryers on a paper machine are typical continuous process heating units. Before the sheet is carried on to the dryers, they have been brought up to temperature. Once a sheet is over the machine, the steam condensed in any dryer will vary but little. In fact, it is not at all unusual for a paper machine to run an entire week on one grade and weight of paper, hence steam pressure and condensate will be uniform. The pressure and amount of condensate will vary between very narrow limits, which is in marked contrast to the wide limits prevailing in typical batch process work. The steam trap that will handle a continuous process job at the highest pressure usually has sufficient capacity to handle all condensate at low pressures. In other words, condensate varies with the pressure in continuous process work instead of inversely proportional to pressure as in batch process work.

In passing, it should be pointed out that when steam is first turned into a continuous process unit, the pressure and condensation curves are similar to batch process curves. However, this condition lasts only until the unit is brought up to temperature, after which the condensation and pressure figures are usually proportional.

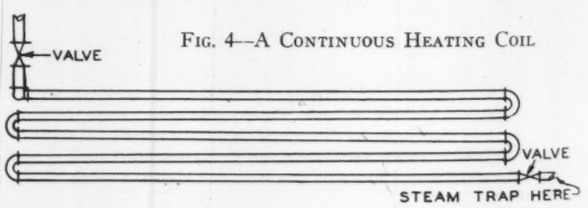


FIG. 4—A CONTINUOUS HEATING COIL

* This is the third of a series of five articles.

Assuming that a distinction has been made between continuous and batch process work, a consideration of the various types of heating units is in order. Although there are many kinds of process or heating units made, a classification based on the type of steam chamber used shows that there are but four types, as follows:

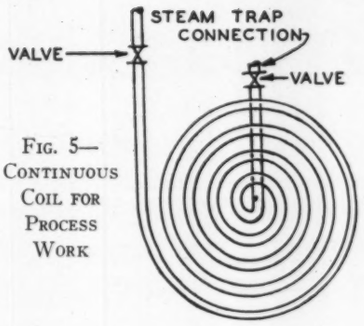


FIG. 5—CONTINUOUS COIL FOR PROCESS WORK

1. Continuous coil type.
2. Chest or chamber type.
3. Siphon drained type.
4. Header coil type.

Fig. 4 shows a very common continuous heating coil used for general heating work. This is made from ordinary pipe and fittings. Fig. 5 shows a type of coil usually made from copper tubing which is used extensively in the food and chemical industries, for example. Heating coils of the type shown in Fig. 4 are used widely for general heating duty (continuous operation) while the coil in Fig. 5 is used usually for batch process work.

A continuous coil heating unit is more easily drained than chest or cylinder type units. The cross-section is so small compared with the total steam space that the steam velocity through the coil sweeps all air and condensate along to the end of the coil and the steam trap. For best results the coil should have continuous gravity drainage to the trap, leaving no pockets in which condensate can accumulate. It is not possible to short circuit any portion of a continuous coil and hence air binding is impossible provided, of course, that the steam trap has sufficient air handling capacity for the particular job.

The jacketed kettle shown in Fig. 6 and the flat work ironer shown in Fig. 7 are typical chest type steam heated process units. As in the case of continuous coils, these chamber type units may be used for continuous or batch process work and a given unit on batch work will require a larger trap than if it were on a continuous process. The steam outlet is at the lowest point of the chamber so that there will be good gravity drainage for

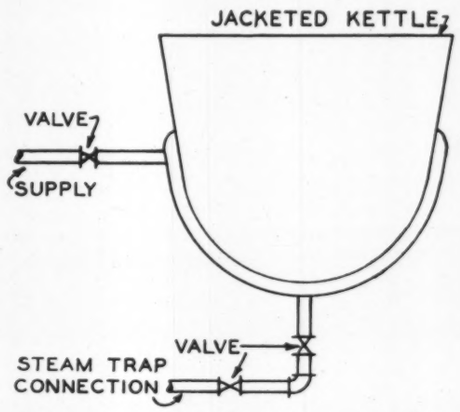


FIG. 6—STEAM TRAP CONNECTION FOR A JACKETED KETTLE

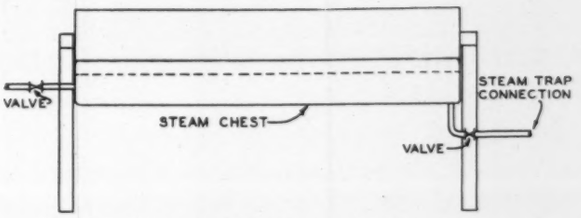


FIG. 7—FLAT WORK IRONER—SINGLE CHEST

condensate. However, the cross section is so large in proportion to the chamber capacity that there is no positive flow to sweep air and other incondensible gases up to the trap. Thus it may happen that these chambers may have "dead" spots in which air can accumulate and cut the heat transfer capacity of the unit. It should be obvious that the air handling capacity of a trap on this type of service is of even more importance than for continuous coil service.

Siphon-drained drying cylinders as shown in Fig. 8 are extensively used on paper machines, cloth slashers, ironing machines, dry milk machines, etc. Steam enters the dryers at *A* and the condensation that collects at the

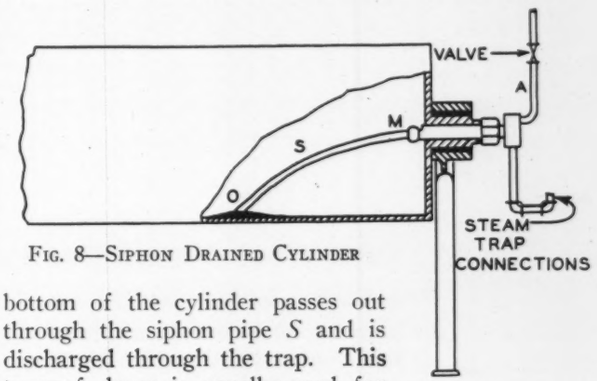


FIG. 8—SIPHON DRAINED CYLINDER

bottom of the cylinder passes out through the siphon pipe *S* and is discharged through the trap. This type of dryer is usually used for continuous process work and hence is brought up to temperature before the process load is applied. Thus the trap does not need much excess capacity over and above the maximum process load.

Draining Cylinder Dryers

Two rather serious difficulties complicate the proper draining of cylinder type dryers, first the large amount of air that is usually encountered and second, the flash steam that is generated in the siphon pipe. When steam is first turned into these dryers they, of course, are filled with air and unless the steam trap used has a large air handling capacity, it is quite likely that all of the air will not be eliminated, even though the dryer be used continuously for a period as long as a week. In addition to the air that is present when steam is first turned on, the steam used contains a certain amount of incondensible gases which will accumulate very rapidly unless positive means are used for removing these incondensible gases as fast as they accumulate.

In order to promote flow through the siphon pipe, the pressure at *O* must be greater than the pressure at *M*. Also, the static head at *O* will be greater than at *M* and this drop in pressure is great enough and quick enough so that considerable flash steam will be encountered at *M*, choking the flow through the siphon pipe itself and also closing the trap before all the water has been dis-

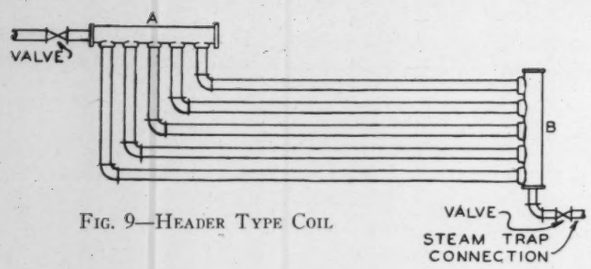


FIG. 9—HEADER TYPE COIL

charged. This flash steam must be condensed or otherwise dissipated, for instance, by special vents, before condensate will flow freely to the trap. There are a number of cylinder type dryers in use that have buckets for collecting the condensate and dumping it into the discharge line rather than siphon pipes. In this type of dryer little trouble is experienced from flash steam as the condensate flows to the trap entirely by gravity and not because of the differential in pressure between the trap and the cylinder itself.

The three types of process heating units just described are all single steam chamber units. Draining the single continuous coil is a comparatively simple matter. Drainage of the chest or chamber type unit is complicated by lack of positive flow to carry air to the trap. The siphon-drained cylinder is even more difficult to handle than the chamber type, both because of the large size of the cylinder (sometimes five feet in diameter and twenty feet long) and because drainage is effected through a siphon pipe. Several single chamber units may be combined into one heating unit or machine. Frequently the steam consumption of one unit of such a machine will be greater than in the other units. This means that the pressure will drop in this particular chamber and very often this pressure drop will cause one or more chambers to short circuit and hold back air or condensate, thereby reducing the heat transfer capacity of the machine.

Multi-Unit Heaters

To bring out the point more strongly, let us consider some multi-unit heaters or machines. The simplest of these is the header type coil, outstanding examples of which are shown in Figs. 9, 10 and 11. Fig. 9 shows a simple type that is commonly used for industrial heating. Headers *A* and *B* may be cast in one piece or made up from ordinary pipe fittings. Fig. 10 shows an angle type that is used in dry kiln work while coils as shown in Fig. 11 are commonly used in rooms for drying clay products.

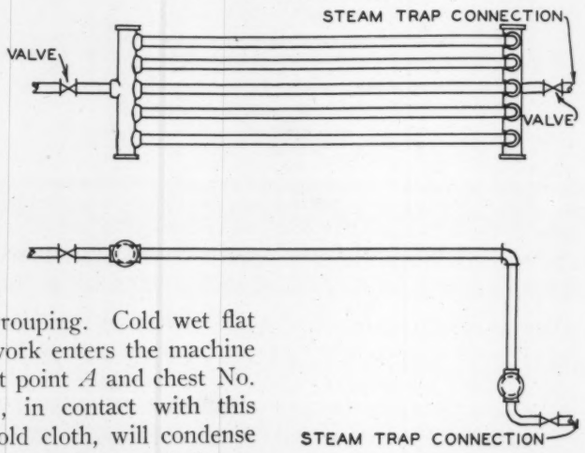
Assume that steam is turned into the coil shown in Fig. 11. The coil is cold and the trap valve wide open. Because of unequal pipe lengths, or because of the fact that some pipes are nearer the steam inlet than others, steam will get to header *B* through some pipes more quickly than others. It is quite probable that steam would reach header *B* through pipes 3 and 4 while air still remains in pipes 1, 2, 5 and 6. By the time steam does reach header *B* through pipes 3 and 4, condensate will reach the trap and before this condensate can be discharged, pressure must build up. As the pressure builds up, air in pipes 1, 2, 5 and 6, is compressed and steam will enter these pipes from header *B*, pocketing the air as indicated by the shaded sections. It is quite probable that these air pockets will remain indefinitely, as steam will continue to flow into pipes, 1, 2, 5 and 6 from header *B*. Condensate from these lines will flow by gravity to header *B* but the flow of condensate towards *B* is not necessarily going to prevent a counter flow of steam and as long as steam is coming in from header *B* the air pockets will remain.

Let it be distinctly understood that this particular layout is a hypothetical case. It is conceivable, but not probable, that steam would reach header *B* through all pipes simultaneously. In this case no air pockets would form. The "steam race" to header *B* seldom ends in a tie and air pockets are likely to form in only one, or all but one of the pipes of a coil of this type. In which coils air pockets will form can best be determined by experiment.

Once these air pockets form, a steam trap can do nothing to remove them. All the trap can do is pass condensate and air that comes to the trap—it can not reach out and pull air against a flow of steam. Coils of this shape should be avoided whenever possible. The fewer, the shorter, and the better drainage of the pipes, the less likelihood there is of air pockets forming. Proper design of steam supply headers to insure uniform flow of steam to each pipe is very important if maximum heat transfer is to be expected from these header-type coils. The ideal way would be to put a trap on each pipe and thus insure positive air and condensate drainage. However, the cost of the installation frequently makes this inadvisable.

Several steam heated chests may be grouped together to make a steam heated machine. A chest type flat work ironer as shown in Fig. 12 is a fine example of this

FIG. 10—HEADER TYPE COIL



grouping. Cold wet flat work enters the machine at point *A* and chest No. 1, in contact with this cold cloth, will condense more steam and hence have a lower pressure than the other chests. The pressure in chest No. 2 will not be as low as in chest No. 1 but will be lower than in the remaining chests. The pressure in the other chests will vary but not to the same extent as the chests on the wet end of the machine. This pressure differential does not interfere with condensate draining as the condensate will flow to the lowest part of the chest and can flow out counter current to any steam that may enter at *M* or *N*. However, as long as the pressures in chests 1 or 2 are even a fraction of an ounce lower than in header *B*, steam will flow from *B* to the chests. This means that air can not escape from the chest until the pressure differential is reversed. As

air accumulates in a chest, there will be less room for steam and hence the amount of condensation will drop and the pressure will rise. Immediately, the flow in pipes *M* or *N* will be reversed, allowing air to escape until pressure again falls off in the chests. That this is not mere theory has been proved by installations where traps have been placed on each chest. The effectiveness of the air removal is proved by increases in the amount and quality of work turned out by the machines.

Twin cylinder flat work ironers and dry milk machines almost invariably have a trap on each cylinder to insure perfect drainage. If a single trap were used, it is highly probable that one cylinder would become short circuited and fail to drain. On the other hand, it is unusual to have individual traps on all paper machine dryers, even though the dryers will range in number anywhere from 20 to 150. These dryers are usually grouped into two sections, wet and dry end; sometimes an intermediate section is provided. The wet sheet from the press section first goes over the wet end dryers which are under low pressure so that the sheet will not dry too rapidly. When the sheet leaves the presses, it is about two-thirds water and at a low temperature, especially during the winter time. Consequently in the first few dryers there is a very large amount of steam condensed and a progressive pressure rise from one dryer to another as the sheet is brought up near steam temperatures. Because of this large amount of condensate and pressure variation, individual traps are often used to drain the dryers in the wet end section. By the time the sheet reaches the dry end or intermediate section, the variation in condensate from one dryer to another is very small—hence there is only a negligible pressure difference. These dryers are usually drained by maintaining sufficient pressure differential between supply and return headers to insure flow through each siphon pipe to the return header. Flash steam, or rather flash steam and live steam, that blows through the dry end dryers is often used as the steam supply to the wet end section.

The above is an extremely brief outline of a paper machine dryer drainage system. There are many patented systems and many arrangements for dividing the dryers into sections and using flash steam from one section in the dryers of the previous section. The point we make is that individual traps are desirable when there is considerable condensate and pressure variation between cylinders—otherwise, one or more dryers will probably become partly filled with water.

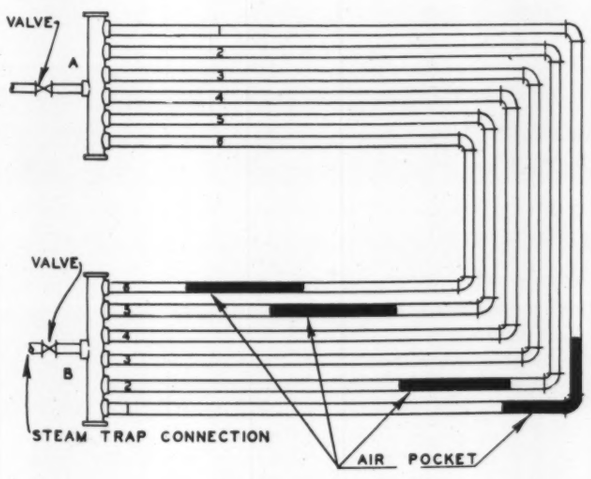


FIG. 11—HEADER TYPE COIL

Analyze any process heating unit and it will be found to be either a continuous coil, chest, or siphon-drained cylinder or some combination of any of the three. Trap selection for any process unit should be compartively simple. If continuous process, the trap should have enough excess capacity so that it will handle the maximum process load comfortably. If a batch process, the trap will have to handle the flood of condensate when starting up and still be able to open the large valve after the pressure builds up and the amount of condensate falls off. Regardless of the service, a trap should be installed that will handle any amount of air that will come to the trap.

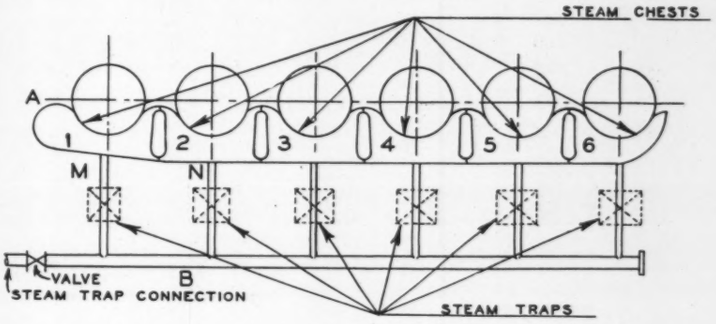


FIG. 12—SIX-ROLL FLAT WORK IRONER

Trap Size Required

The maximum steam consumption and hence the size of trap required to drain any apparatus can be estimated.

TABLE 1—STEAM CONDENSED PER FT. OF BARE PIPE PER HR. AIR TEMP., 75 F

Gage Pressure	Steam Temperature	Temperature Difference	Pipe Diameter in Inches									Per Sq. Ft. Large Pipes or Flat Surfaces
			1	1½	2	3	4	5	6	8	10	
5	228	153	.16	.21	.25	.35	.45	.54	.64	.82	1.01	.34
50	298	223	.28	.38	.45	.63	.79	.97	1.14	1.47	1.80	.62
100	338	263	.36	.41	.59	.84	1 05	1 28	1.51	1.94	2.39	.82
160	366	291	.43	59	.71	1.00	1.26	1.54	1.82	2.35	2.89	.99

Where the radiation is in greenhouses, dry kilns, etc., the condensation figures should be doubled, while for drying wet materials such as brick, pottery, etc., triple the above figures. For indirect heating coils with blower fan, the condensation will be from five to six times as great and in the coils on the cold air side, the figures may be increased by another fifty per cent. On the other hand, if the lines are well insulated, the condensation figures can be divided by five.

The condensation from coils or other surfaces covered with water depends upon the metal used and the difference in temperature between the steam and the liquid. Table 2 shows the condensation to be expected under such conditions.

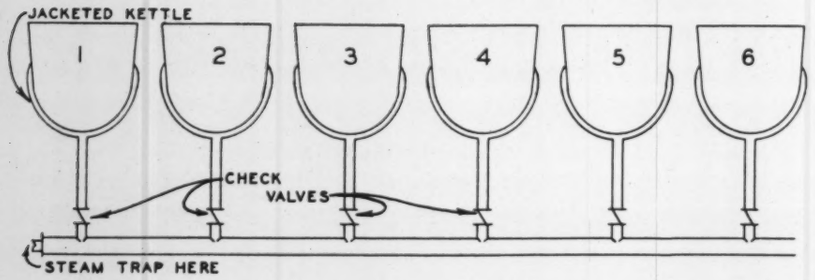


FIG. 13—SIX JACKETED KETTLES DRAINED BY ONE TRAP

TABLE 2—CONDENSATION FROM STEAM COILS SUBMERGED IN WATER

TEMPERATURE DIFFERENCE BETWEEN STEAM AND WATER DEG. F	POUNDS OF STEAM CONDENSED PER SQUARE FOOT OF SUBMERGED SURFACE		
	WROUGHT IRON COILS	BRASS COILS	COPPER COILS
50	10	16	18
100	32	50	58
150	66	100	114
200	110	164	186
300	220	330	380

The above table applies to metal surfaces that remain clean and if the surfaces become fouled, the condensation will be reduced. Condensation will be further reduced if the liquid contains solid matter. For viscous liquids, the condensation figures may be anywhere from one-fourth to one-eighth of the values in the above table.

After the probable condensation from a unit has been determined, the selection of a steam trap is a simple matter. From the trap manufacturer's capacity tables, pick out a trap that has the capacity necessary plus enough excess capacity to handle any extraordinary operating conditions. If, after the trap is installed, it apparently lacks capacity, a test should be made with the next size larger traps. In general, it is comparatively easy to estimate the size of trap for draining coil radiation but for chest or chamber type units, larger traps than the theoretical are almost always necessary.

Just as continuous coils, steam chambers, and siphon-drained cylinders are combined to make single units usually drained by a single trap, two or more machines may also be drained by a single trap. There are many successful installations of this type. On the other hand, there is a very great possibility that one or more machines will short circuit and become filled with air or water and thus go out of commission. Fig. 13 shows six jacketed kettles drained by a single trap. As in the case of the header type coil, steam will reach the header more quickly through some kettles than others. Thus steam will pass from the return header to the kettle, blocking off the escape of air. Suppose check valves are installed in the outlet lines from each kettle so there can be no back flow of steam. Let us further suppose that the material in kettle 3 is dumped and a batch of cold material added. Immediately the pressure in kettle 3 drops, the check valve closes and no condensate can escape until the pressure in kettle 3, together with the static head of condensate on the check valve, is equal to the pressure on the discharge header side of the check valve. Until the pressure in kettle 3 is equal to the pressure in the header no air can escape as there will be a water seal over the check valve. This means that condensate and air will cut the heat transfer capacity of kettle 3 as long as the pressure in the kettle is less than the return line pressure. If a trap were installed on each kettle, there would always be a drop in pressure between kettle and return line, hence air and condensate would always be free to pass through the trap. Individual trapping costs more than group trapping. However, first cost is not final cost and final cost determines the profit. One batch of material spoiled by slow cooking will frequently pay for an individual trap — not to mention loss of time and overhead. Hence, modern engineering practice does not ask "How cheaply can we trap these units?" rather, "How can we trap these units in order to get the largest output of high grade material?" The cost of a trap is very low, often less than 1 per cent of the cost of the heating unit. Is it logical then to attempt to economize on 1 per cent of the investment and cut the capacity of 99 per cent of the investment anywhere from one-tenth to one-half?

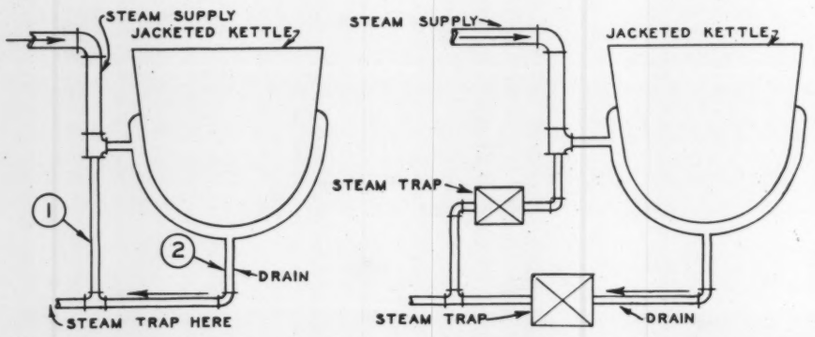


FIG. 14—INCORRECT MANNER OF TRAPPING A JACKETED KETTLE (LEFT). CORRECT WAY (RIGHT)

Sometimes engineers who insist on individual traps for steam heated units will attempt an installation such as shown in Fig. 14. Line No. 1 drains the supply header while line No. 2 drains the kettle. A pressure approximately equal to header pressure will build up in the trap through line No. 1. The steam condensed in the kettle will keep the pressure in the kettle well below header pressure and consequently the kettle will not drain satisfactorily. This hook-up is capable of many variations and should be avoided. It is better not to drain the supply line than to short circuit the unit. Of course, sound engineering would insist on a trap to drain the header and a separate trap to drain the kettle.

The fourth of this series will appear in an early issue.

Air Conditioning a Department Store

By E. H. Dafter

THE GREATER HUDSON STORES OF DETROIT

TO THE J. L. Hudson Co., Detroit, Michigan, belongs much of the credit for pioneering the application of air conditioning to department stores. In April, 1923, the management of the J. L. Hudson Company, realizing that its expanding business was outgrowing its quarters, decided to enlarge their quarters by replacing the original wood construction building, erected in 1891, with a new fireproof building of the finest type for its purpose.

Not content with mere increase in floor space, the J. L. Hudson Company aspired to the finest merchandising and mechanical equipment of the day. When this new building, now known as the Farmer Street Bldg., was proposed, the Hudson Company requested its architects and engineers to make a special survey and study of the heating and ventilating requirements with particular regard to the ventilation problem of the basement store. After intensive study and investigation the architects and engineers suggested air conditioning instead of mere ventilation. The result was that the main floor, the mezzanine and the two basement floors of the new eighteen-story building erected in 1924 and 1925 were air conditioned.

Some time after the original installation was completed, and during the construction of the company's second new building a few years later, more of which will be told later, the management of the Hudson Company was being interviewed in regard to the air conditioning installation and during the conversation the following remark was made, "We consider air conditioning absolutely essential for our basement store. In order to sell merchandise below the ground level, one of the first requisites is a *first class* ventilation system. Without adequate ventilation it would be impossible to conduct sales in the basements and operate a first class store with any degree of success. We believe that in the long run air conditioning does not cost us any more than an adequate ventilation system would for this reason: Floor space is a valuable item in a department store and means dollars and cents. With air conditioning we save space because we can install smaller equipment, smaller ducts, etc., as compared with that which a ventilation system might require. There are also operating economies that can be effected due to the fact that we can recirculate a large part of our air. As to air conditioning our main floor, we charge that off to advertising. We realized that if we could make our patrons comfortable and draw them in off the street, the equipment would soon pay for itself in the advertising that results therefrom and also the additional sales that may result."

The author believes that the above statement will prove to be of considerable interest to those in the air conditioning field.

Growth of the Hudson Company was rapid and the history of the company is one of almost constant expansion. Early in 1927 the business of the Newcombe-Endicott Co. was purchased and in October, 1927, razing operations were started on the buildings just purchased, to make room for the new building which houses the second installation of air conditioning equipment previously mentioned. The management of the Hudson Co. was so completely satisfied with air conditioning that it was one of its major considerations in the new structure now known as the Newcombe-Endicott Building.

Conditions Maintained

The equipment designed and installed in each of the Hudson Co. buildings was specified and is guaranteed to maintain 74 degree dry bulb and 55 per cent relative humidity inside the store, when the outside does not exceed 81 degrees dry bulb and 73 degrees wet bulb.

The same equipment that provides the cooling and dehumidifying in the summer also provides the heating and humidifying in the winter.

Winter conditions are maintained at 72 degrees

AN INTERIOR VIEW OF THE FARMER STREET ENTRANCE TO THE FARMER STREET BUILDING. NOTE THE AIR SUPPLY GRILLE ABOVE THE REVOLVING DOORS

and 40 per cent relative humidity. It is interesting at this point to note from proved operation that very often when the outside temperature stood at 32 degrees or below the basement store has required cooling instead of heating; that is, the temperature of the air delivered to the store was lower than that withdrawn. This is the result of the tremendous amount of heat given off by the customers and employes and from the electric lights. In fact, during unseasonable weather in the holiday season when the thermometer occasionally reached 50 degrees or above, it has been necessary to operate the refrigeration units in conjunction with the air conditioning units.

Location of Equipment

The air conditioning plant is located in the third basement level. The first installation is located in the Farmer street building and consists of five sets of air conditioning units, and three refrigeration machines of the centrifugal type. Air conditioning unit No. 1 is designed to take care of the main floor and mezzanine. Units No. 2 and No. 3 condition the first and second basement sales areas. Unit No. 4 is designed to condition the third basement stock storage room and also the machinery room, while unit No. 5 was arranged to connect into the existing ductwork of the south portion of the existing Woodward avenue building.

The second installation is located in the Newcombe-Endicott building and is also in the third basement. It consists of eight sets of air conditioning units and three units of the centrifugal refrigeration machines. Air conditioning unit No. 21 conditions the entire main floor and mezzanine of the Newcomb-Endicott building. Units No. 22, No. 23, and No. 24 are designed to take care of the first and second basement sales areas. Unit No. 25 conditions the north portion of the old Woodward avenue building. It is so arranged that it takes care of the main floor and mezzanine, first basement sales area, and also the general offices of the basement store which are located in the second basement. This portion of the old building was revamped and remodelled during construction of the Newcomb-Endicott building. Unit No. 26 conditions the third basement stock storage and receiving room. Unit No. 27, the fourth basement packing and shipping departments, while unit No. 28 takes care of the cooling and ventilating of the machinery room. There is also a small heating fan unit installed in this room which is capable of delivering air at 150 degrees to each entrance vestibule on the main floor. These vestibules have an air change approximately every two minutes and are designed to reduce infiltration losses in the winter time.

Capacity of Equipment

The Farmer street building has a total capacity of 192,000 c.f.m. combined with 600 tons of refrigeration handled in the air conditioning plant. The second plant, which was completed early in 1929, added 350,000 cu. ft. of air per minute and 1,000 tons of refrigeration. The refrigeration machines in the first installation are driven by three 200-hp., 3600-r.p.m., 220-volt, 3-phase, 60-cycle, constant-speed squirrel-cage induction motors. The machines in the new installation are driven by three 350-hp., 1800-r.p.m., variable speed, slip ring motors with 4600-volt, 3-phase, 60-cycle current.

The outdoor air is drawn in by fans at the tower floors through neatly grilled windows and is immediately tempered and filtered. It is then passed down the elevator shafts under a unique system of temperature and pressure control. This is accomplished without the creation of drafts and no discomfort is experienced by elevator riders. Finally, the air is delivered to each air conditioning unit in the third

basement. Here, in the summer time, it is dehumidified, cooled and washed in the sprays of water circulated from the refrigerating units. It is then mixed under automatic control with a quantity of recirculated air which has previously been filtered and is finally discharged through an extensive duct system to the conditioned areas. The entire plant is under automatic control and the air in the conditioned spaces is held within a two-degree variation.

System of Ductwork

One of the interesting features, and also one that demanded considerable thought and attention on the part of the designing engineers, was the arrangement of the ductwork. In general, all trunk ducts were run on either the third or fourth basement ceilings, with the fourth basement having the majority. There is approximately 172 tons of sheet metal ductwork suspended from this ceiling. From the trunk ducts on these ceilings a myriad of small branch risers were tapped off and run up vertically to supply and exhaust from the individual grilles.

Air Distribution

The air distribution of the main floor is through grilles arranged around the outer walls and over the show cases. Various trunk risers were run up at advantageous points and discharged into specially designed sheet metal plenums that were built into the upper part of the show cases. This point was worked out very carefully and to detail with the manufacturer of the show case fixtures. These grilles discharge outwardly approximately eight to ten feet above the floor level. For the inner central portion of the main floor, a few branch ducts were run in the false ceiling of the first basement and arranged to discharge up vertically between show case fixtures placed back to back. This air was sprayed out into the room much in the same manner of water issuing from a fountain. The recirculated and exhaust air is drawn in at the floor line through specially designed grilles underneath the show case fixtures.

The supply air for the mezzanine is carried up in the outside walls in small branch risers from the trunk ducts on the basement ceiling and is carried out into false beams and discharged horizontally through wide shallow grilles on either side of the beam. No return air is taken from the mezzanine but is allowed to spill over onto the first floor, where it is drawn back into the system.

The air for the first and second basement sales area is delivered in a unique and interesting manner. Both the supply and exhaust are distributed through very small ducts built around each column and which in turn are plastered in and formed as part of the column. This method avoided the use of unsightly overhead ductwork and left the already restricted headroom entirely clear, since in both new buildings the first and second basement levels had to match up to the existing levels of old buildings. As mentioned previously, the trunk ducts for these risers were run on either the third or fourth basement ceilings, which are not sales areas open to the general public. This method of arranging the ducts as part of the columns presented some very interesting problems, which were worked out in close co-operation with the consulting and structural steel engineers.

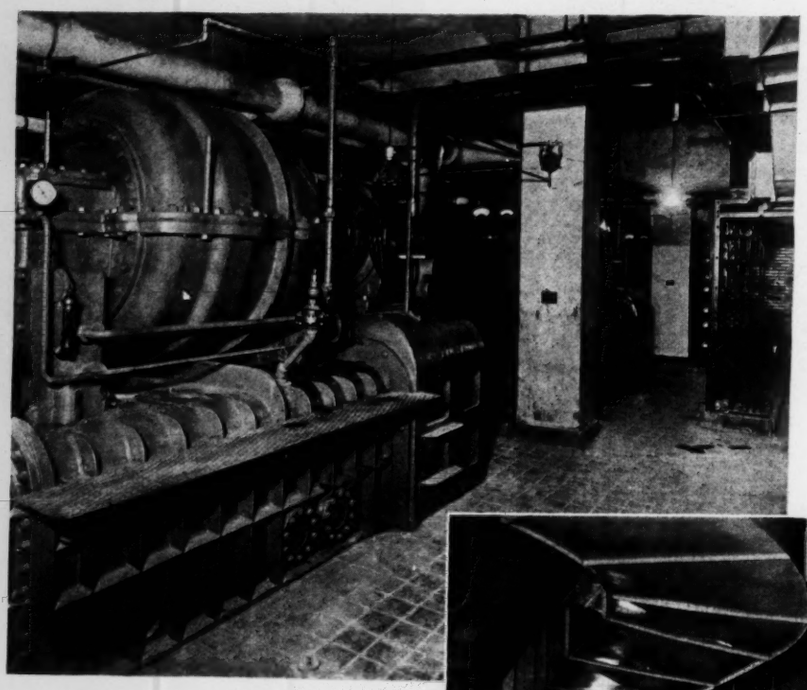
The air distribution of the third and fourth basements is of conventional design and needs no special comment. The air is delivered through diffuser outlets in the ducts at the ceiling level and exhausted at the floor level in all cases.

Automatic Control

In the summer time, automatic thermostats control the dewpoint or saturation temperature of the air leaving the spray chambers. Individual automatic thermostats are placed in the various recirculating ducts from each floor or division of a floor and react upon mixing dampers which control the ratio of dehumidified and recirculated air. Since the dewpoint temperature is predetermined and regulated, controlling the temperature in the conditioned areas naturally maintains the relative humidity desired within a very small percentage, allowing for the moisture given off by the people.

In the winter time, the dewpoint thermostat functions in the same manner as before, but set at a lower temperature, while the thermostats in the recircu-

A Glimpse of One of the Conditioned Dining Rooms on the Thirteenth Floor. The Ornamental Ceiling Air Diffuser Is Combined with the Lighting Fixture

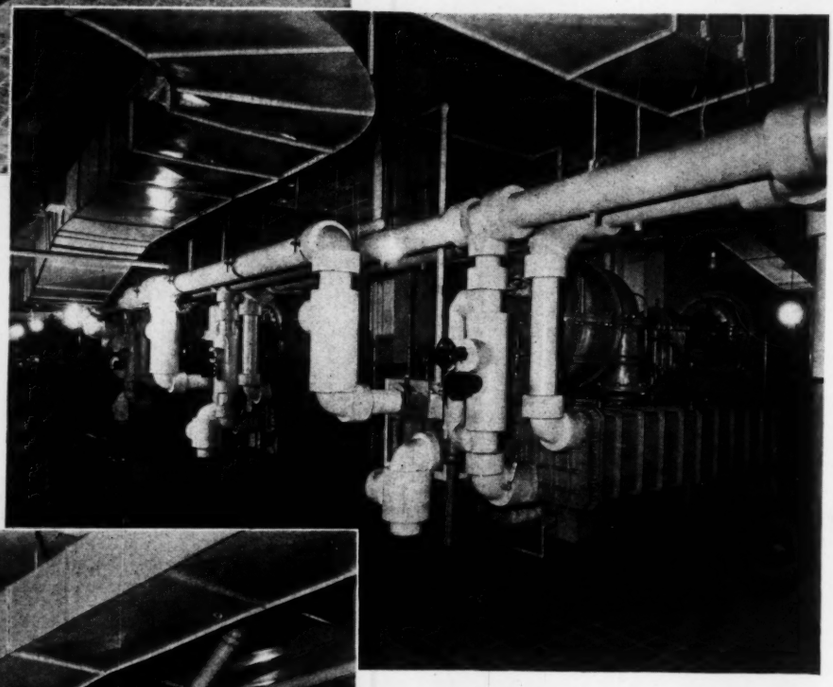


One of the Three 200 Ton Refrigeration Units Located in the Third Basement of the Farmer Street Building of the Hudson Stores. Each of These Complete Units Occupies a Floor Space of About Ten by Fourteen Feet

lating ducts react upon diaphragm steam valves in addition to the dampers. The diaphragm valves control the admission of steam to heaters which are placed in the path of the recirculated air.

Recording Instruments

In order to provide the operating engi-



(Right) Three of the Centrifugal Refrigerating Units in the Recent Newcombe-Endicott Addition. The Capacity of Each of These Units Is 330 Tons. The Unit in the Foreground Shows the Water-Cooler on the Left, the Refrigerant Condenser on the Right, While the Motor and Compressor Are Mounted on These Casings as a Base

A View of the Fans, Spray Chamber and Starting Controls in the Newcombe-Endicott Addition. Winter Conditions Are Maintained at 72 Degrees and 40% R. H.

neers with a continous record of the functioning of the air conditioning plant, four electrical eight-point recording instruments of the continuous type were installed. This allows the recording of the temperatures at thirty-two various points and one can tell at a glance just what is occurring at each recording station. These instruments were connected up in this manner. Machine No. 1 was so arranged that it recorded eight dewpoints only, one from each conditioning unit. Instrument No. 2 recorded nothing but return air temperature, one station being at each of the eight conditioning units. Machine No. 3 recorded eight fan discharge temperatures, while instrument No. 4 recorded various temperatures as follows: (1) fresh air dry bulb; (2) fresh air wet bulb (3) chilled water to sprays; (4)

chilled water return from sprays; (5) city water supply to refrigeration condensers; (6) condenser water discharge; (7) transformer room exhaust temperature, and (8) machinery room exhaust temperature.

Additional Air Conditioning

In addition to the air conditioning plant for the sales area just described, the Hudson company has two other air conditioning plants that are of interest. One of these is the plant for the air conditioning of the public restaurants located on the thirteenth floor and the other is what the author believes is the first air conditioning plant for the cooling and ventilating of the transformer vault in a building of this nature.

The Transformer Vault Air Conditioning Plant

Air conditioning was installed here for the primary consideration of space saving requirements. The Hudson company has one of the largest, if not the largest, privately owned transformer vaults in the state of Michigan and the ventilation of this vault had to be given special consideration and treatment due to space limitations for ventilation equipment and fresh air supply and exhaust risers. The common method of ventilating such areas is to bring untreated or unconditioned outdoor air into the vault, allow it to circulate through the vault and finally discharge outdoors after it has absorbed as much of the heat losses as possible. The consulting engineers and architects demonstrated to the owners that, if air conditioning, combined with refrigeration and evaporative cooling were adopted, the equipment could be reduced approximately to one-third of that which would be required for ventilation alone.

This equipment is designed and guaranteed to maintain less than 100 degrees in the vault, which is always below the specified limited surrounding temperature of 104 deg. fahr. at which point the transformers begin to decrease in efficiency. The supply system has a capacity of 20,000 cu. ft. of air per minute while the exhaust system is rated at 24,000 cu. ft. per min.

The supply system consists of fan, dehumidifier, filter and preheater, but no reheater, as no attempt is made to control any definite temperature in the vault other than to keep it below 100 deg. fahr. The outdoor air is drawn in from the street level, is filtered, washed and cooled in the spray chamber and is then delivered through a specially designed duct system, the outlets of which are arranged so as to direct the air against each transformer in order to obtain the maximum cooling effect of the air where desired. The supply duct in the vault is built upon the floor and discharges the air at the floor line while the exhaust is removed through fiber grilles in the ceiling.

The automatic control on this unit is a very simple one and is designed to effect all of the operating economy possible as regards the saving of refrigeration load. It consists of a graduated-action thermostat placed in the exhaust duct which operates a pneumatic pressure switch. This switch in turn controls the starting and stopping of the motor driving the pump supplying refrigerated water and also a three-way diaphragm valve on the suction of the spray water pump. Normally, all of the cooling required is obtained by evaporative cooling and refrigeration is not required. Under these conditions the spray water pump recirculates the water from the tank of the spray chamber, through the sprays and cools the air passing through the dehumidifier to its entering wet bulb temperature. The operation continues in this manner until the temperature in the exhaust duct reaches 95 degrees, at which point the thermostat in the exhaust duct starts up the refrigerated water pump, which delivers cold water to the three-way valve on the suction of the spray pump. This valve admits only the necessary amount of cold water to the sprays in order to keep the exhaust temperature at 95 degrees. As a safety device there is also an electrical thermostat in the vault which rings an alarm bell in the engineer's office in case the temperature should ever reach 100.

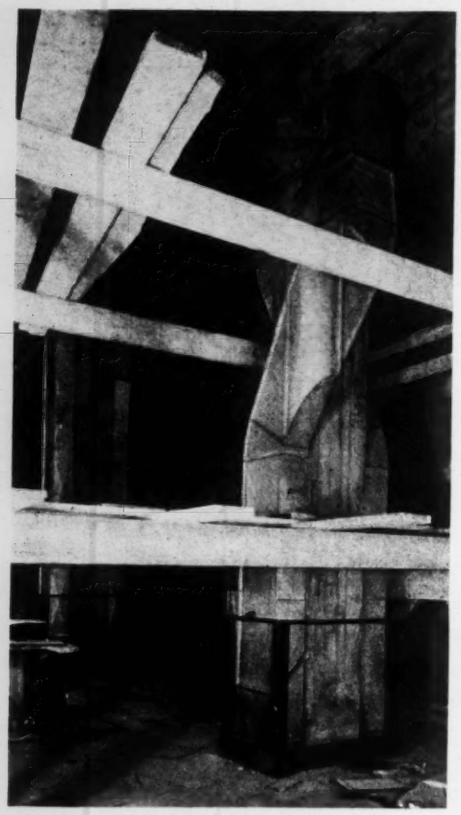
Dining Room Air Conditioning

The entire thirteenth floor of the Hudson store is devoted to public dining rooms, which are air conditioned from a plant designed for these rooms and which is entirely independent from any other air conditioning system in the building. This area is divided into three separate and distinct dining groups; the Early American Group, which is subdivided into three rooms, each of which has an individual architectural treatment, but all are connected by open archways; the Cosmopolitan group, which consists of one large room and a number of small individual private dining rooms, at either end of the large room; and the men's grill which, as is denoted by the name, is treated architecturally and furnished in an appropriate fashion. Dividing this off into groups of dining areas and further subdividing it complicated the air conditioning design, since each one of these areas had to be treated so that it could be operated independently and made to respond immediately to meet wide variations in occupancy.

The air conditioning apparatus was placed in a pent house on the roof of the old Woodward avenue building and was located on a level slightly below the fourteenth floor of the new building. In this pent house were located all the fans, preheaters, washers, air filters, reheaters, etc. The refrigeration unit, for cooling the spray water in this system, was placed in a machinery room on the eighteenth floor level. It is also of the centrifugal type and has a refrigeration capacity of 200 tons and weighs 40,000 lb. It is driven by a 250 hp., 1800 r.p.m., variable speed, slip ring motor wound for 220 volts, 3 phase, 60 cycle current. The air conditioning plant has a total capacity of 62,000 c.f.m., outdoor air. It is so designed that only outdoor air can be taken through the dehumidifier, though the supply fans can take a certain amount of recirculated air which by-passes the dehumidifier, but is independently filtered and treated with ozone, the latter for the purpose of eliminating food odors.

The arrangement of the apparatus is slightly different from that conditioning the sales areas in that there is only one common dehumidifier from which

three individual supply fans draw conditioned air for the respective dining areas. Space limitations required the use of only one dehumidifier while the use of three supply fans gave greater flexibility and ease of operation in maintaining the three dining groups at the proper temperature and humidity. It was considered that there may be times when only



THIS PHOTOGRAPH SHOWS THE MANNER IN WHICH THE SMALL BRANCH RISERS WERE INSTALLED AROUND THE COLUMNS AS THEY PASS THROUGH THE THIRD BASEMENT. NOTE THE MANNER IN WHICH SOME RISERS HAD TO TRANSFER FROM ONE SIDE OF THE COLUMNS TO THE OTHER IN ORDER TO MEET THE STRUCTURAL STEEL REQUIREMENTS

any one, or possibly two, of the main groups would be required and at that time the proper fans could be shut down. When the variable occupancy of each area is considered as to its effect upon the air conditioning requirements, the use of three fans seems highly justified even though space limitations required the use of one dehumidifier.

To simplify further the ease and operation of control, the supply ductwork for both the Early American dining rooms and the Cosmopolitan group is divided into main supply headers for the different rooms. These rooms are under automatic control from a room thermostat controlling a volume damper in the supply duct. The dehumidifier has its usual dewpoint control, while a thermostat in the recirculating duct functions in the same manner as on the units in the sales areas. The ductwork is run in false ceilings above the dining rooms. The supply air is delivered through specially designed ceiling diffusers which harmonize with the architectural treatment and are so disguised that the layman would not know the point of air delivery. The exhaust and recirculated air is removed at the floor line through unobtrusive grilles in the walls around the dining rooms. The exhaust air in this case is delivered to the kitchens as part of the kitchen supply while the recirculated air is drawn back to the air conditioning units to be used again in order to temper the supply air.

Kitchen Ventilation

There are two large kitchens in connection with these dining rooms which, while not air conditioned in the strictest terms, required special consideration in the heating and ventilating equipment. One of the kitchens is designed primarily to serve the Early American group of dining rooms while the other is designed to serve both the men's grille and the Cosmopolitan group of dining rooms. There is, in addition to the supply air already mentioned as being exhausted from the dining rooms into the kitchens both directly and indirectly, a large fan which draws air from outdoors, through filters and tempering surfaces and thence delivers it through special drop outlets in the ceiling of the kitchens. The total supply to both kitchens is 70,000 c.f.m. producing a two-minute air change. For winter time operation there is a thermostat in the fan discharge controlling the diaphragm valve on the tempering heater. It was found, in actual operation, that if this thermostat were set at 60 degrees it produced comfortable conditions in the kitchen. The supply fan was located on the fifteenth floor level.

The exhaust fan, which is located on the seventeenth floor and discharges its air out through grilled windows on the eighteenth floor, has a capacity of 90,000 c.f.m. This gives slightly less than a two-minute air change in the kitchens and, since it has a greater capacity than the total supply, produces a negative pressure in the kitchen which prevents any cooking odors from getting out into the dining rooms. To nullify any odors which may be discharged outdoors through the exhaust and perhaps re-enter the building through an open window, a high concentration ozone machine was installed to kill such odors as they were generated, before being discharged outdoors through the exhaust. About 90 per cent of the exhaust air is withdrawn from the kitchen through hoods over the ranges, steam tables, dishwashers, etc. All of the exhaust duct was constructed of 14-gage, black iron and insulated with 2 in. of fire felt insulation.

Insulation of Cold Water Pipe

Dripping of power plant piping caused by sweating is being given more attention. It may be taken care of by using anti-sweat insulation, which is made in several forms. One of these is made in two layers of insulating felts with waterproof felt applied next to the pipe between the two layers of felt insulation, applied with staggered joints, and on the outside of the insulation.

Piping for Economy and Service

By B. M. Conaty

PRACTICALLY every large industrial plant, and hundreds of smaller ones, have problems in plant operation or maintenance involving the use of steam, water, oil or other gases or liquids. In a great number of these cases, large economies may be effected through the use of piping systems if specially designed to meet the requirements.

The problem may be one of steam utilization for heating and power purposes. Frequently we see large industrial plants which have grown up spasmodically with little opportunity for intelligent planning of plant extensions, and economical arrangement for steam generation and distribution. One frequently sees such plants with two, three or even more boiler plants within a radius of a few hundred or a few thousand feet. The tendency today is to concentrate all steam generation at one modern plant, or to utilize the full capacity of one of the existing plants, adding such boiler capacity as may be necessary to replace all of the other separate boiler plants. The steam requirements of each individual building, then, are supplied from the central station through specially designed underground or aerial high pressure steam transmission lines, and tremendous economies are thereby secured.

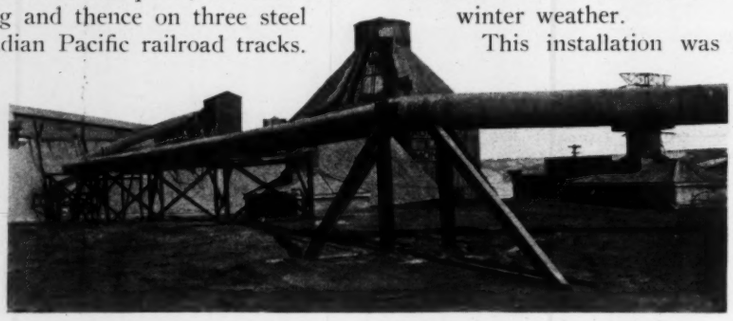
The story of the major piping installation completed within the past year by the Shawinigan Chemicals, Limited, of Shawinigan Falls, Quebec, is of particular interest. In addition to its carbide manufacturing plant, this company has an acid plant, both located on the banks of the St. Maurice River. The two plants are approximately 3500 ft. apart and were originally operated quite independently.

With every outlook indicating a considerable increase in demand for its products, the company planned a considerable plant extension program for both the carbide and chemical division during 1928. In connection with the new lime kilns for the carbide plant, waste heat boilers were installed to generate steam at 175 lb. pressure and 100 deg. superheat. There was a need for this steam for process work in the chemical plant. The problem was to transmit the steam approximately 3,500 feet with minimum line loss and pressure drop. The engineers decided on an 8-in. high pressure welded line.

Extra strong steel pipe was selected so that considerably higher pressures might be employed later should this be desirable, at any time, to increase the carrying capacity of the line. By reason of the unusual local conditions rendering the adequate anchorage of expansion joints extremely difficult, line expansion was accommodated by the use of special pipe loops. The entire line was insulated by the application of 3-in. thick 85 per cent magnesia sectional covering surrounded with a 1-in. layer of hair felt. The insulation was then protected with a heavy waterproof roofing jacket. The jacket was held firmly in place by means of copper weld wire ties closely spaced and all joints were carefully sealed to make them waterproof.

The installation started at the boilers, the line was carried through the roof trusses and entire length of the kiln building, thence on steel bridges and bents to the

carbide grinding building, along one wall of this building on steel brackets and trusses, over the gable roof of one intervening building and two 35-foot gaps to the roof of the can plant. From this point, the line continued over this building and thence on three steel bridges across the Canadian Pacific railroad tracks. From this point down to the acid plant, a distance of about 3,000 feet, the line is carried on concrete piers or steel posts provided with cast iron roller chairs. The balance of the line within the acid plant is again above ground, carried on steel pipe columns and rollers to the various plant buildings where connections to equipment are made.



One interesting feature of this installation is the fact that the only fabricated material shipped to the job were the steel bridges crossing the Canadian Pacific tracks and the straight lengths of pipe. All other steel structures, supports, etc., were produced on the ground by means of welding, bending, etc. The expansion loops and changes in direction comprising seventy, long radius, extra heavy pipe bends of various degrees were all made on the job to fit the conditions encountered. The entire installation was carried out in the course of the severest winter weather.

This installation was subjected to extremely severe tests before its acceptance, the first test consisting of a 1,000 - pound hydrostatic pressure with the impact test applied at each weld simultaneously. The steam test consisted of turning the boiler pressure into the line suddenly by opening the valve at the boiler header just as fast as a man could do so. This occurred when the surrounding temperature was in the neighborhood of 40 degrees below zero.

The tests above described developed no defects of any description and steam has not been off this installation since it was put into service during the early part of 1929. An important phase of plant modernization is piping for greater economies.

Air Conditioning for Diseases

By Charles Bryne

A SEARCH of the medical literature, with a view of learning the proper climatic conditions needed to fight disease, is a discouraging venture. Not that the literature is scant on the subject. In fact it is too full of conclusions arrived at by review of only a few cases or by results undoubtedly controlled by many variables outside of those noted. Furthermore, the interest which some doctors have had in the treatment of certain diseases in a particular region has tended to cloud the true issues. As a result, one will find much conflict in the opinions given. For an example, patients with laryngeal tuberculosis are advised to go to mild damp climates, to dry sunny climates and sometimes to the mountains.

An outstanding conclusion, which has stood the test of experience for many years on the part of the medical profession, is the discovery, by Dr. Loewy, that there is a marked increase in blood pressure as the elevation above sea level increases and that this high blood pressure can be relieved by the respiratory administration of oxygen. This statement does not mean that the medical profession, as a whole, accepts in full these two conclusions, but that modern practice is swinging more and more to these views. In more recent years, Dr. Huggard has gone further than Dr. Loewy. He has shown that the red cells of the blood increase as the altitude increases. The reason for this peculiar adjustment, which also reverses its action as altitude decreases, appears to be an effort of nature to maintain the supply of oxygen in the lungs. Thus, changes in the partial pressure of oxygen in the outside atmosphere, which would vary the oxygen inhaled, are compensated.

One of the earliest medical investigators to appreciate the separate and combined effects of air motion, dry bulb temperature and relative humidity on the human body was Dr. Leonard Hill. He concluded that the higher cooling power of outside air increase the metabolism during exposure and raises the basal metabolism to a higher level. For the benefit of the engineer, who can not be expected to understand the language of the medical profession, the definition for metabolism is given. It is "the sum of the processes concerned in the building up of protoplasm and its destruction or the chemical changes, in living cells, by which energy is provided for the vital processes and activities and new material is assimilated to repair waste."

As early as 1920, Dr. Egbert Morland saw the practical application of the research carried on by Dr. Hill. He decided that "personal climate," or the climate within one's clothes, was more vital than outside climate, that damp air was not damp to dry-clothed people who had warm skins, that the skin is the agent through which the greater part of the effect of climate is transmitted to the organisms and that personal climate, as far as the lungs are concerned, begins at the back of the nose. Here is the basis through which the true relations between various diseases and their treatment by artificial weather, or air conditioning, can be uncovered. Dr. Morland pointed out that a layer of air, in contact with the warm skin, has its relative moisture reduced to a very low

point if the skin is dry, that the skin must have a stimulating environment, that such environment could be supplied by properly controlled air motion and radiant heat, that pigmented skin deals more safely and easily with all manner of climatic influences and that pigmentation of the skin could be obtained even in the diffused light of a clouded sky where wind or rain are present.

If air conditioning has any value in the treatment of disease, this information should give a foundation on which to build. To attempt to apply the medical conclusions outlined above, in a practical manner, is the purpose of this paper.

In order that radiant heat may be used to the best advantage, it is the opinion of the writer that hospitals should be provided, in connection with air conditioning plants or otherwise, with radiators which are efficient in delivering radiant heat. Such radiators, for gravity service, may be found in steam pipes coiled into radiators, cast iron radiators of the smaller types, single column cast iron radiators, wall type cast iron radiators and the newer fin radiators made of sheet metal. This list is not complete nor is it arranged in the order of efficiency. As the various forms or types of fan blast radiators on the market are efficient in the delivery of convected heat, it would seem to the writer that the best heating arrangement would be radiant heat delivered from gravity types of equipment reinforced by warmed air delivered from blast radiators, as part of the air conditioning equipment, circulated from the ducts at low velocities around 100 to 200 F.P.M. With the newer type of gravity fin radiation, used in a gravity system alone, the temperature differences between floor and ceiling will be very small, room heights can be lowered and fuel costs reduced.

It is apparent, also, that patients with lung, nose or throat diseases require a higher effective temperature (that temperature which determines the personal degree of comfort for each particular human being under various conditions of dry bulb temperature, relative humidity and air motion) for comfort than those who are not thus afflicted. It also appears evident that the value of air conditioning must be very great for diseases of this character from a dust elimination standpoint alone.

Treatment of some forms of pneumonia, and possibly other forms of lung diseases, with oxygen administration through the nose or mouth has come into considerable use in hospital practice. One great disadvantage has been the fact that this form of administration has been very painful and is somewhat expensive. Where the room is conditioned, oxygen can be introduced into the air in sufficient quantities to obtain the same effect and the patient will never know that the treatment has been given. Furthermore, the cost per patient for this form of treatment is lowered.

Air conditioning equipment installed in any hospital should be flexible enough to provide varying degrees of air pressure as well as a fairly wide range of effective temperatures, dry bulb temperatures and relative humidities. The oxygen treatment outlined above avoids the need for a variation of air pressure in the treatment of lung, throat and nose diseases but asthma, as an example, can be relieved by adjustment of the air pressure.

The relative humidity most suitable is likely to depend upon the disease and also upon the effective temperature required. It appears evident that a variation of from 10 to 20 per cent may be satisfactory in many cases. The pleural effusions of pneumonia, which increase the body discomfort and raise the body temperatures to the fever point, can be relieved by a relative humidity of from 15 to 25 per cent. Coughing also requires the same treatment by a low humidity. Like the effective temperature factor, the relative humidity factor needs considerable study and experiment on the part of the hospitals. Furthermore, there is a technique in the measurement of relative humidity, or relative vapor pressure, which needs to be understood to avoid errors in observations and to put data on a reliable basis. This art will be found in detail on pages 394-395 of the September HEATING, PIPING AND AIR CONDITIONING.

Where the heart is laboring, the kidneys are not functioning properly or the digestion is feeble, every additional strain, due to an imperfect climate or other discomfort, renders a cure less possible. It is not unreasonable, therefore, to see in air conditioning a solution to many hospital problems. The skin can be put in a more healthy condition by radiant heat, the metabolism can be greatly improved, the many discomforts and difficulties incident to different diseases can be eliminated, fever can be effectively reduced and moisture effusions can be rapidly removed. Not only should air conditioning give a larger percentage of cures, but the usual time to effect a cure should be reduced with the use of such equipment.

The cost of air conditioning in hospitals is not likely to be high. This, like many other cost problems, depends greatly on the type of equipment selected and on the accessories provided. No cut and dried figures would mean much until the decision was made as to the spaces which were to be conditioned and as to the flexibility of the equipment desired.

CINCINNATI GENERAL HOSPITAL, WHERE STUDIES ARE BEING MADE ON THE CONTROL OF AIR ENVIRONMENT

Temperature and Humidity Control of an Oxygen Chamber

By John B. Hashagen

THE use of oxygen as a stimulant and aid has long been recognized in the treatment of pneumonia, respiratory and heart afflictions. There are several ways of administering oxygen to the patient but the two most commonly employed are the oxygen tent and the oxygen chamber methods.

When the oxygen tent is used, the head and chest of the patient is enclosed in a rather large tent of transparent and air tight material into which oxygen is introduced. This method, however, does not provide so great a degree of comfort for the patient as may be secured by the use of the oxygen chamber.

To secure the best results, it is essential that the patient have the proper amount of oxygen and be kept in an atmosphere which is maintained at the proper temperature and humidity, all in correct diffusion and circulation. After many years of experiment and research work the conditions have been met by Dr. Alvan L. Barach, a specialist in the treatment of these cases, who has devised the oxygen chamber now in use in the Columbia Presbyterian Hospital, New York City.

Features of the Oxygen Chamber

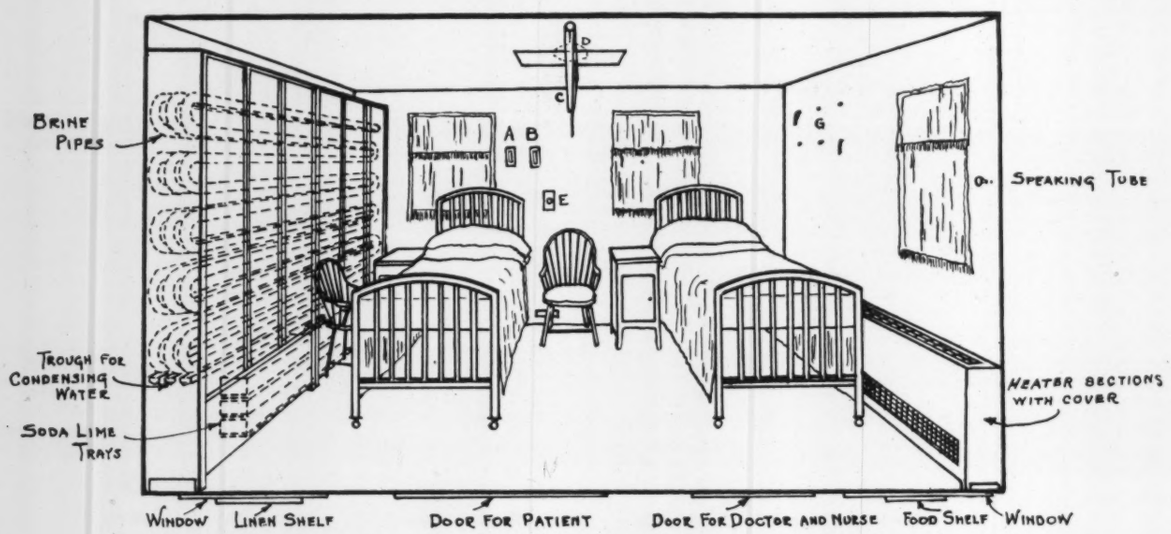
The entire chamber is constructed of aluminum plates, the floor being inlaid with magnesite, and despite windows, doors, etc., is air tight. It is transportable and can be set up in a large room in a hospital, its dimensions being approximately 15 ft. long, 10 ft. wide and 8 ft. high, large enough to accommodate two beds. In the center of the ceiling and over each of the two beds in the chamber are small glass windows through which electric light is supplied when required. Suspended a few inches below the ceiling, and midway between the end walls, is a roll curtain which extends the entire width of the chamber and which may be utilized to divide the chamber into two individual rooms.

Brine Coils

Evenly distributed over one end wall inside the chamber are brine coils of about 900 linear feet of 1-in. brass pipe, a few inches in front of which is a sheet-metal baffle extending from side wall to side wall and to within 6 in. of the ceiling and 12 in. of the floor.

Underneath the coil is a V-shaped trough in which all water drippings from the brine coils are collected and conveyed outside the chamber through a water-sealed pipe.

Brass screen trays filled with soda-lime are also placed beneath the coils and under the trough to absorb carbon dioxide.



OXYGEN CHAMBER AT THE COLUMBIA PRESBYTERIAN HOSPITAL IN NEW YORK CITY. *A* INDICATES THE HUMIDOSTAT; *B*, THE THERMOSTAT; *C*, CURTAIN TO SEPARATE CHAMBER; *D*, CEILING LIGHT; *E*, BUZZER FOR NURSE; *G*, INLETS FOR OXYGEN AND FOR CO_2 AND O_2 TESTS

Heating

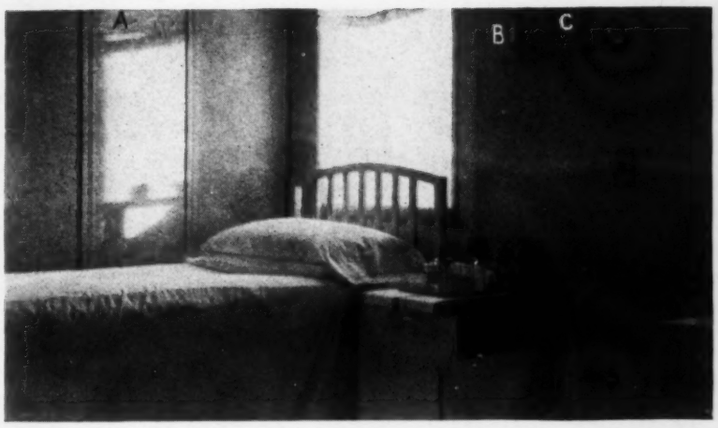
On the opposite end wall of the chamber two small heating sections are placed near the floor and shielded by metal radiator covers with grille work on the bottom for entry of chilled air and on top to aid in giving directional flow to the heated air.

In this wall there is a curtained window. To the right of this window is a speaking tube, while to the left and near the top of the wall are several pet cock inlets for

CO_2 and O_2 tests and for the admission of oxygen from the storage tanks outside the chamber.

The rear wall has two curtained windows evenly spaced. Between these windows and about 5 ft. 6-in. above the floor are a pneumatic room-type humidostat and thermostat.

The front wall has, at the left, a small window underneath which is located a trap door large enough to admit linen which may be passed through from outside the chamber. Then come two doors, the left one being slightly larger than the right in order to permit a bed to be wheeled through. The door on the right is for the use of physicians and nurses. On the right is another window located in a position corresponding to the window on the left and having a similar trap door to admit food.



A View of the Oxygen Chamber Showing Baffle (*A*), Humidostat (*B*), and Thermostat (*C*)

Inside this wall and immediately in front of each of the two large and two small doors are suspended white rubber curtains which aid in preventing escape of oxygen from the chamber when the doors are opened.

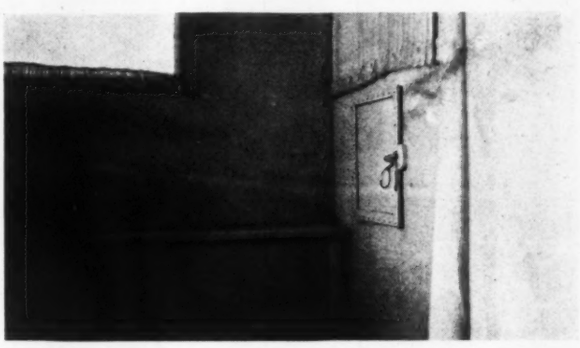
The operation of the chamber is comparatively simple but is productive of highly beneficial results.

The thermostat and humidostat heretofore mentioned accurately control and determine the temperature and humidity in the chamber.

Temperature and Humidity Control

The thermostat operates a diaphragm valve outside the chamber in the steam supply line leading to the heating units placed along the wall opposite the brine coils, and has a range of 50 F.

The humidostat operates a diaphragm valve also outside the chamber in the brine supply line and regulates



Food Is Passed Into the Chamber Through a Small Door

the amount of brine flowing through the coils, thereby fixing the amount of moisture condensing on the coils. No mechanical methods are employed to raise the moisture content of the air. The only moisture available is that given off by the chamber occupants.

Maintaining Proper Conditions

The thermostat and humidostat settings are made as directed by the physician from time to time.

The oxygen content of the chamber is increased by opening the pet cocks above referred to until the desired concentration level, usually between 40 and 60 per cent, is obtained.

The air in this chamber is kept in motion only through the cooling effect produced by the brine coils at one end of the chamber and the heating units at the base of the other end. The air coming into contact with the brine coils is chilled and deprived of its moisture, falls directly to the floor and passes to the heating units where it is drawn through the grille at the bottom and is heated. It then passes up to, and along, the ceiling over to the brine coils where the circulation process is repeated. Air velocity tests were made using a kata thermometer and showed a velocity of 35 ft. per min., 1 ft. from the floor at the brine-coil end, while at other points in the chamber the air had a velocity of from 15 to 20 ft. per min

Good diffusion and circulation are obtained within the chamber as the table of tests below indicates.

Atmospheric Conditions in Chamber with Thermostat Set at 76 F and Humidostat at Dry

		Brine End	Center	Steam End
Ceiling	Dry bulb temperature	79	76	78
	Wet bulb temperature	57	55	57
	Relative humidity, per cent	22	22	24
4 ft. 6 in. from floor	Dry bulb temperature	77	75	74
	Wet bulb temperature	56	55	55
	Relative humidity, per cent	23	24	26
Floor	Dry bulb temperature	61	57	57
	Wet bulb temperature	48	45	44
	Relative humidity, per cent	35	35	31

Results secured by the use of this oxygen chamber in the treatment of pneumonia patients have been so gratifying that many hospitals throughout the country adopted and installed the chamber as a necessary adjunct.

The Effect of Moisture Content on Yarns

The tensile strength of yarns is quite dependent upon the moisture content. The relative humidity and temperature of the test room, during a test, should be 65 per cent and 66-70 F. For the practical comparison of yarn strengths, the temperature and moisture conditions should be constant in the testing room. This can be obtained if the samples are exposed for 24 hours in the room.

Drying Problems in the Manufacture of Telephone Cable

By Malcolm Tomlinson

ANY discussion of the problems connected with the drying of lead-covered telephone cable may well be introduced, for the sake of clarity, with a brief survey of those fundamentals which underlie all drying problems. Drying is concerned with the deposition of moisture, decomposition of water, chemical precipitation, absorption, adsorption, mechanical separation, diffusion and evaporation. The last two methods may be brought under one head — vaporization. In all of these methods, drying is also concerned with equilibrium.

Lead-covered telephone cable drying should be of interest, because it not only involves both the atmospheric and the vacuum methods of drying, but also new and unique methods for holding at a minimum the regain of moisture in dried cable cores.

In his well known statement of the laws governing equilibrium, Willard Gibbs starts by regarding a system as possessing three variables—pressure, temperature and concentration or solubility. He then goes on to show that all systems in true equilibrium may be composed of solid, liquid and vapor phases. When changes occur in temperature, pressure or concentration, the physical characteristics and phase relations of any system are varied. As an example, at 19.5 deg. fahr., silver nitrate has a solubility of 46.2 parts of dry salt in 100 parts of solution, but at 359.5 deg. its solubility has reached 96.9. Again, under atmospheric pressure, water boils at 212 deg. while under an absolute pressure of ½ lb. per sq. in., its boiling point has been depressed to approximately 84 deg.

Although at present most drying work is conducted under atmospheric pressure it is evident that, since changes in physical characteristics and phase relations can be brought about by variation of pressure, the effect of such variation should be studied in any drying problem. This involves consideration of the effects of pressures below atmospheric (or vacuum), atmospheric pressure, and pressures above atmospheric. It also follows that the use of vacuum or of pressures above atmospheric offer, in combination with changes in temperature and concentration, more variations than can be obtained if the pressure is held constant, or practically constant, as when atmospheric pressure is employed.

The use of vacuum drying has been developed greatly in recent years due to the fact that the color and the delicate physical structure of many materials, which are seriously affected at 212 deg. fahr., are not harmed if the boiling point of water can be sufficiently depressed. Recently, in the paper industry, considerable commercial success has been secured by using pressures above atmospheric in the beater rooms, but it is probable that this method of drying has had very little consideration in other fields where it appears suitable.

Lead-covered telephone cable drying should be of interest because it not only involves both the atmospheric and the vacuum methods of drying but also new and unique methods for holding at a minimum the regain of moisture in dried cable cores.

Extent of the Lead - Covered Cable Industry

In considering any problems involved in the manufacture of lead-covered telphone cable it is interesting to keep in mind the magnitude of the product handled and the rate of increase by which this volume has been attained.

In 1901 the Western Electric Co. used 2,131,000,000 feet of wire in the manufacture of telephone cable. By 1910 this had increased to 7,706,000,000 and in 1917 to 13,432,000,000. In 1925 it had increased to 36,000,000,000 and in 1929 to 49,000,000,000 feet. In drying this amount of cable, over 600 tons of water were removed.

A study of the problem of drying paper-insulated, lead-covered cable makes it necessary to consider many factors other than simply the removal of moisture from the paper insulation. The reason for this is that, so far, no commercial means have been developed for determining the quantity of moisture remaining in the paper after the telephone cable has been dried. The efficiency of the drying operation is still determined by the electrical characteristics of the finished cable. These characteristics are affected by at least twenty-five other conditions which are all variable. Thus, the effect of a single variable condition is extremely hard to determine. An investigation of this kind requires that a large number of cables must be tested. In running a series of tests, to obtain the effect on the finished product of varying a given condition, all of the other variables must be held as nearly constant as possible and a sufficient number of tests must be run so that the uncontrolled variations in any of the processes not under consideration will be averaged out of the final results.

Two Types of Telephone Cable

There are two general types of paper-insulated, lead-covered telephone cable. One is the subscriber's loop cable used to connect the subscriber's telephone to the central exchange and also to connect local exchanges. The other is the long distance or toll cable used in connecting towns or exchanges that are some distance apart.

At present it is possible to telephone from Boston to Chicago entirely over circuits incorporated in lead-covered telephone cable.

As the equipment for drying telephone cable is of the most modern design and arrangement, it would be interesting for any drying or air conditioning engineer to visit such installations. While only loop cable is manufactured in some plants the processes involved in the manufacture of toll cable are, in general, similar, except that greater care is used in the co-ordination of the

A Sheathing Machine Used in Making Cable

circuits and that the drying equipment for toll cable is more elaborate in order to obtain special results.

Two Processes in Drying Loop Cable

The general processes involved in the manufacture of loop cable are as follows:

The bare copper wire is insulated by wrapping with a paper ribbon. The width and thickness of this paper ribbon vary depending upon the type of conductor and the characteristics desired in the finished cable. Two conductors are twisted together to form a single telephone circuit. A number of twisted pairs of wires are stranded together to form the cable core. The number of pairs placed in a single core may vary from 6 to 2,424. The stranded cores are fed on special trucks from the stranding machine and, after being tested for mechanical defects in the circuits and repaired, are dried and then covered with a lead sheath. When the cable is cool, following the lead covering operation, it is given the final electrical tests and is then ready to be crated for shipment.

Factors Affecting Electrical Characteristics

The variable items in the construction of telephone cable, which directly affect the electrical characteristics of the cable, in general are the character of the insulating paper, the tightness with which the paper has been applied to the wire, the tightness with which the wires are twisted together and the tightness with which the pairs are stranded into the cable core, the efficiency of the drying equipment and the tightness of the lead sheath on the cable core. Some of these variables can, to some extent, be controlled in the manufacturing processes while others cannot. Some are a question of raw material, some are dependent wholly or partially upon atmospheric conditions and others are dependent upon the skill of the operators and on the machine adjustments.

There are numerous variables directly affecting the rate of drying of cable cores which roughly may be grouped under three headings; raw materials, construction of the cable core and efficiency of the drying equipment.

Variation in raw materials used in the manufacture of the insulating paper will affect its physical properties. The density of the paper, its variation in thickness and stiffness resulting from changes in the calendaring process and the arrangement of the fibers in the paper, all affect its characteristics and vary its response to any drying process. The effect of varying these relations may be seen in papers published by Hertzberg, Veitch and Reed, Kress and Silverstein and by Houston, Carson and Kirkwood.

The amount of air space, also the amount of copper in the cable as well as the tightness and regularity with which the paper wrap is applied to the copper conductors, have a marked influence upon the characteristics of the finished cable and upon the rate of drying (the rate at which the moisture in the insulating paper can be expelled). The humidity at the time of insulating, twisting and stranding greatly affects the above operations as the higher the relative humidity the greater is the tendency of the paper to wrap tightly on the wire. Also, the higher the relative humidity, the greater the tendency of the wires to lie close together in the twisting and stranding operations. The tension on the wires during these operations also influences the tightness of the cable core. The tension applied to the supply reel is controlled on the machines performing these operations.

Factors that Influence Rate of Drying

The rate of drying is influenced not only by the condition of the paper, the tightness with which it is applied to the conductors, and the tightness of the cable core, but also by the amount of cable on the trucks, the absolute pressure in the vacuum driers, the temperatures in the vacuum driers and the moisture in the insulating paper at the time the cable is placed in the vacuum driers.

The dryness of the cable core, when lead-covered, depends upon its dryness at the end of the drying period, the regain in moisture while the cable is still in the drier due to the introduction of moisture in the air used to break the vacuum and the regain of moisture during the period in the storage oven after drying and before lead covering. In connection with the latter variable, regain, the paper by Kress and McNaughten on the "Regain of Papers at Different Humidities" contains useful information. The moisture content of the paper insulation at the time of making the final electrical tests is determined by the dryness of the core at the time of lead covering and the adsorption of moisture in the ends of the cable during the testing operation. The influence of this latter item is controlled by variations in the prepara-

tion of the ends of the cable for testing and the relative humidity in the room during testing operations.

The foregoing points give an idea of the variable conditions with which it is necessary to deal in determining the most economical cycle for the drying process.

The Drying Equipment

The equipment usually consists of cylindrical driers of the horizontal type (although vertical driers are used in some plants), surface condensers with connected receivers and reciprocating types of dry vacuum pumps, reinforced with steam jet pumps where very low absolute pressures are desired. The vacuum driers are heated by steam coils placed longitudinally in the driers. The ends of the driers are closed with doors which are equipped with lifting devices so that the door may be raised and swung into the open or closed position. Some driers are equipped with a secondary outlet, in addition to the one for the vacuum connections, in order that circulation of air may be had during the heating-up period prior to the vacuum period. Where low humidities are used in the storage oven, the individual driers are connected by a small pipe to the oven so that a minimum of moisture will be introduced when the vacuum is broken. The condensing and pump equipment is cross-connected to the two mains, one to raise and one to hold the vacuum, and the driers are each connected, through a single vacuum outlet, to both mains. This type of layout requires many valves but is most flexible.

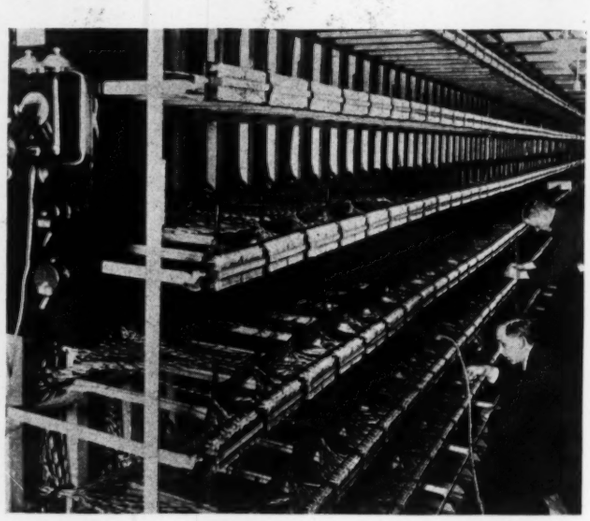
The storage oven is usually located close to the vacuum driers. Here the cable cores are held until they can be covered by the adjoining lead presses and, as this storage period lasts from 15 minutes to 24 hours, every effort must be made to maintain the condition of the cores as to moisture content so that regain can be kept at a minimum. This condition usually involves low humidities around 10 per cent with air temperatures of from 110 to 120 deg. fahr. The vacuum driers sometimes extend into this oven for unloading purposes and the cable core is fed to the lead press through an opening in the wall.

Dehumidification

Dehumidification equipment used to condition such rooms consists of air washers, centrifugal or reciprocating compressors using CO_2 in the latter case or dielene in the former instance, condensers and evaporators. The air motion is slow, about 100 ft. per min., and the ductwork must be laid out so that the air is uniformly distributed. In order that adjustments may be maintained it is necessary to take readings of the relative humidity with some degree of accuracy with the aid of an aspirating psychrometer of the indicating type. Automatic controls are needed to govern the admission of outside and room air to the washers. Recently improvements have made possible the use of chemical dehydration as an auxiliary to this refrigerating equipment. Brine is usually required in the air washers. In such cases the gaskets used should be of a high grade material which will not be affected by brine. Calcium chloride is an economical and satisfactory freezing agent for such purposes. It has the disadvantage of heavy entrainment in the air and will, due to this fact, gum up piping and "weep" at some joints. Sodium chloride is more corrosive to the metal in the washers and piping, does not entrain to an appreciable extent but has a cyrohydric point which makes it objectionable for continuous operations where dew points are any lower than 35 deg. fahr. In other words, sodium chloride will solidify into a mass which resembles ice at a temperature around 17 deg. fahr. for concentrations of sufficient strength for operation purposes. This does not furnish sufficient leeway below the dew-point and may cause shutdowns at undesirable times.

The Drying Cycle

The drying cycle on paper insulated telephone cable is about as follows: cable cores are placed in the vacuum drier and the doors are closed and locked. Steam already has been turned on so that the interior of the drier is quite warm. A so-called heating up period of from one to three hours is used prior to the application of vacuum.



General Divisor, Horizontal Side of a Modern Telephone System

The steam temperature is usually held at about 250 deg. fahr. although it is not good economy, as a rule, to use such high pressures needed to obtain this temperature. When the vacuum is applied the steam temperature drops and a higher pressure is needed, temporarily, to bring the temperature back to normal. It usually requires from 5 to 15 minutes to obtain a vacuum of 1 lb. per sq. in. absolute. A similar period will be required to drop to a pressure of about ¼ lb. per sq. in. absolute with the steam jet pump. The time the cable cores are held under a vacuum will depend greatly upon the pressure maintained, but for 1 lb. per sq. in. absolute, this period is usually from 6 to 9 hours. Another factor which influences the vacuum period required is the thickness of the cable core on the core truck. The average cycle between the time the core enters the drier and the time it is sheathed is about 16 hours.

The drying of toll cable by vacuum should be accomplished at lower absolute pressures than are used for loop cable or for a longer vacuum period.

In the storage oven, toll cable requires a relative humidity not higher than ½ per cent at temperatures from 120 to 130 deg. fahr. This condition, as well as the condition given for the storage of loop cable cores, will furnish an atmosphere which is somewhat warm but still comfortable.

Heat Transfer Constants *and* Applications of Heat Transfer

By W. R. Woolrich and L. Holdredge

FIVE general applications of heat transfer are encountered in heating and ventilating engineering. These are, (a) heat transmitted from the room or building to the outdoors through windows, walls, ceilings, floors and doors; (b) heat transmitted through or from the heating devices from the heating mediums to the rooms; (c) heat transmitted through insulation provided to decrease the heat transfer at points where heat is not required; (d) heat transfer from the burning fuel to the heating medium; and (e) heat transfer from the heating mediums to process liquids and materials.

In this article the authors have given constants which have been developed from the best available data given for the unit transfer rate, then recomputed for a 70-degree difference in temperature and rearranged to be more usable to the engineer and contractor.

A number of values have been changed by the writers when they discovered that these values were inconsistent. In every case, however, they have made use of such constants and have found them to work out consistently in their consulting engineering practice.

The basic unit from which most of these values were derived was secured from the unit per degree value as given in the A. S. H. V. E. Guide.

Methods of Heat Transmission

Heat may be transmitted in three ways: Convection, conduction and radiation. In any given case, two or even all three may be present.

In convection, some medium is heated at the source of heat. Thus by convection, the Gulf stream carries heat from the Gulf of Mexico to western Europe. In the same way the water next to the bottom of a tea kettle is heated and rises to the surface. Currents of air which flow from warm to cool places also carry heat. Convection occurs when a substance moves from one place to another of different temperature.

Conduction is the movement of the heat along and within the body itself. Put the end of the poker in the fire and it becomes hot. The heat travels along the poker and the other end is warmed. This is conduction. If some other substance is brought into contact with a heated body this substance will be heated by conduction of heat from one body to the other.

Radiation is the transfer of heat from the hot to the cold body without the heating of the intervening substance. Radiation heat travels in straight lines and obeys the same laws as light. The sun warms the earth by radiated heat. In our heating systems, conduction and convection are always present with radiation.

Fallacies and Variables in Heat Data

Early heat transmission experiments proved conclusively the great value of dead air space. Unfortunately, great errors have been made by zealous advocates of some products, when such products gave evidence of enclosed air compartments.

Dead air space is effective only when convection currents are not present. With porous materials and poorly made joints, convection currents are ever present. The chimney effect of the space sometimes aggravates or increases the activity of the convection currents to make the so-called "dead air space" a most active heat carrier.

Experiments and experience show that the most effective dead air spaces are very small, thus reducing the opportunity for air currents to exist.

The variables most pronounced in heat transmission data are those of materials and construction methods. Thus, any table on the heat transfer rates through brick walls must necessarily contain an appreciable factor of ignorance, since the texture, mortar, joint, and course arrangement may vary greatly, and a variation of any one or all of these factors would appreciably affect the transfer rate of heat through the wall. Likewise, any other building material is subject to so many variations in construction methods employed that any tabulated data must be used with discretion.

The Heating Problem

The problem of heating a building resolves itself, therefore, into (1) a determination of the heat requirements for such room or building to care for heating the incoming air from outside temperatures to those required within; (2) the calculation of the heat loss through the walls, ceilings, doors, windows and floors, which heat must be replaced by the heating system if any given temperature is to be maintained, and (3) a determination of what supplementary heating may be anticipated from auxiliary devices, equipment and animal life within the building.

Heat Requirements for Incoming Air

This factor requires some careful estimates. From the weight of a cu. ft. of air at normal room tempera-

tures, it is found that it takes .02 B.tu to raise the temperature of one cubic foot of air one degree F. One B.tu will raise 50 cu. ft. of air one degree F under average conditions.

But the air changes per hour produced either by natural convection or air infiltration, or by forced air circulation involve more indeterminate values. Table 1 gives the average air changes per hour required for natural infiltration or convection for different types of rooms and buildings. However, when forced circulation is contemplated, these values are usually much too low. Rooms are not air tight and cooler air will enter the room around the doors and windows. Additional air must be supplied for ventilation if many people are in the room. If the room is not continuously heated, heat must also be supplied to bring it up to the required temperature.

There is sometimes a tendency for salesmen of forced circulation equipment to recommend a large number of air changes in order to sell larger fan and circulatory equipment. If the air is recirculated, which can be safely done when properly conditioned, the heating requirements may be kept reasonably low. But an over-ventilated room using cold outside air for the fan supply can be a very expensive luxury. Industrial processes which involve dangerous or undesirable fumes and therefore require these high rate changes for the protection of life and property are a notable exception to this statement.

TABLE 1—AVERAGE AIR CHANGES PER HOUR FOR VENTILATION

KIND OF ROOM OR BUILDING	AIR CHANGES PER HOUR
Halls	3
First floor rooms	2
Offices on first floor	3
Upper floor rooms	1–1½
Upper floor offices	2
Auditoriums	1–2
Stores	2–3
Factories on natural circulation	1
Factories on forced circulation	2–15
Textile mills	3–4
Restaurants	6–10

Problem: How much heat would be required to raise the temperature of two changes of 10 F air to a room temperature of 70 F, the room being 40 ft. by 80 ft. by 10 ft.?

Computation:

Number of cubic feet in room will be

$$40 \times 80 \times 10 = 32{,}000$$

Since the air must be heated from 10 F to 70 F, the temperature increase will be 60 F.

For two changes it will require then $2 \times 32{,}000$ or 64,000 cu. ft. of air to be raised in temperature 60 F.

It requires 1 B.tu to heat 50 cu. ft. of air one degree.

To raise one cubic foot 60 F will require $\frac{60}{50}$ or 1.20 B.tu.

To raise 64,000 cu. ft. of air 60 F will require 1.20 × 64,000 or 76,800 B.tu. *Answer.*

Heat Requirements for Heat Transmission from Buildings

If a room is at a higher temperature than its surroundings, heat will flow from the warm room to the cooler surroundings. If the room is to be maintained at this higher temperature, heat must be supplied to replace that lost by conduction through, and radiation from, the walls, windows, doors, ceiling, floor, etc.

The material out of which the walls, etc. are made enables us to calculate the amount of heat required to raise the temperature any given number of degrees, assuming average values as given in Tables 3 to 12.

TABLE 2—INSIDE TEMPERATURES FREQUENTLY REQUIRED

KIND OF BUILDING	DEGREES F
Residences	70
Auditoriums	68
Stores	67
Schools	69
Offices	69
Hospitals	70–75
Greenhouses	60–85
Bath Rooms	75–85
Factories for light work	65
Factories for heavy work	60

TABLE 3—HEAT LOSS THROUGH BRICK WALLS

Heat loss for exterior brick walls in B.tu per hour per square foot for 0 F outside and 70 F inside with a wind velocity of 15 miles per hour. For northern and western exposure, add 10 per cent to these values.

CHARACTER OF WALL	THICKNESS OF WALL				
	4 INS.	9 INS.	13 INS.	18 INS.	24 INS.
Plain brick with mortar	43	27	19	11	9
Plain brick with mortar and plastered	34	22	17	11	9
Plain brick with mortar, plastered and furred	25	16	14	10	8

TABLE 4—HEAT LOSS THROUGH CONCRETE WALLS

Heat loss from exterior concrete walls in B.tu per hour per square foot for 0 F on outside and 70 F on inside with a wind velocity of 15 miles per hour. For northern and western exposure, add 10 per cent to these values.

CHARACTER OF WALL	THICKNESS OF WALL			
	4 INS.	8 INS.	12 INS.	16 INS.
Plain concrete	53	35	28	24
Plain concrete with 4-in. brick face	31	26	21	18
Plain concrete with 4-in. brick face, plastered inside	28	21	17	15
Plain concrete with 4-in. brick face, furred, lathed and plastered inside	21	18	15	12

Adapting Tables to Other Temperature Conditions

Tables 3 to 10 are based on an outside temperature of zero F, and an inside temperature of 70 F. The outside temperature may be assumed at from 15 to 20 F above the lowest temperature recorded by the *United States Weather Bureau* at the nearest station. The lowest recorded temperature at Knoxville, Tennessee, is 16 F below zero; Washington, D. C., 15 F below zero; New

TABLE 5—HEAT LOSS THROUGH HEAVY STONE WALLS

Heat loss from exterior heavy stone walls in B.tu per hour per square foot for 0 F on outside and 70 F on inside with a wind velocity of 15 miles per hour. For northern and western exposure, add 10 per cent to these values.

CHARACTER OF WALL	THICKNESS OF WALL					
	4 INS.	8 INS.	10 INS.	12 INS.	16 INS.	20 INS.
Plain stone, rustic mortar joints	85	75	68	60	55	50
Plain stone, heavy cement mortar	65	50	44	40	35	33
Plain stone, heavy cement mortar, plastered inside	55	45	40	35	28	22

TABLE 6—HEAT LOSS THROUGH HOLLOW TILE WALLS

Heat loss in hollow tile walls in B.tu per hour per square foot for 0 F on the outside and 70 F on the inside with a wind velocity of 15 miles per hour. For northern and western exposures, add 10 per cent to these values.

CHARACTER OF WALL	THICKNESS OF WALL		
	4 INS.	6 INS.	8 INS.
Hollow tile	38	30	24
Hollow tile, plastered inside	32	26	20
Hollow tile with 4-in. brick face	26	22	18
Hollow tile with 4-in. brick face, plastered inside	20	17	15
Hollow tile with 4-in. brick face, furred, lathed, and plastered inside	16	14	12
Hollow tile, furred and plastered inside, stucco outside	24	20	16
Hollow tile, plastered inside, stucco outside	29	23	18

TABLE 7—WOOD WALLS

Heat loss from wood walls in B.tu per hour per square foot for 0 F on outside and 70 F on inside with a wind velocity of 15 miles per hour. For northern and western exposure, add 10 per cent to these values.

Wood siding on studs	48
Siding on studs, lath and plaster	34
Siding on studs with sheathing, lath and plaster	26
Siding on studs with paper, sheathing, lath and plaster	21
Siding on studs, brick veneer, lath and plaster	21

TABLE 8—HEAT LOSS THROUGH DOORS AND WINDOWS

Heat loss in B.tu per hour per square foot for 0 F on the outside and 70 F on the inside with a wind velocity of 15 miles per hour. For northern and western exposures, add 10 per cent to these values.

Windows, doors, and skylights—loose	90
Windows, doors, and skylights—well installed	78
Windows, doors, and skylights, with weather strip	60
Windows, doors, and skylights—double with air space	52

TABLE 9—HEAT LOSS THROUGH ROOFING MATERIAL

Heat loss from roofs in B.tu per hour per square foot for 0 F on the outside and 70 F on the inside with a wind velocity of 15 miles per hour. For northern and western exposures, add 10 per cent to these values.

Shingles on wood strips	61
Shingles on wood sheathing	30
Shingles on strips over paper and sheathing	15
Flat tile on strips	75
Flat tile on sheathing	46
Slate on strips	77
Slate on sheathing and paper	31
Tin on wood strips	111
Tin on sheathing	41
Tin on paper and sheathing	30
Corrugated iron on strips	107
Corrugated iron on sheathing	50
Felt roofing on sheathing	25
Paper, tar and gravel on 1½ sheathing	18
3 in. concrete, paper, tar and gravel	44

TABLE 10—INTERIOR WALLS

Heat loss in B.tu per hour per sq. ft. for 35 F on the unheated side and 70 F on the heated side.

Studs with lath and plaster on one side	21
Studs with lath and plaster both sides	12
4-in. hollow tile plastered one side	18
4-in. hollow tile plastered both sides	16
2-in. gypsum blocks plastered one side	21
2-in. gypsum blocks plastered both sides	19

TABLE 11—CEILINGS AND FLOORS NEXT TO UNHEATED SPACE

Heat loss in B.tu per hour per sq. ft. for 35 F on the unheated side and 70 F on the heated side.

Lath and plaster on joists	21
Single wood floor on joists with lath and plaster ceiling including finish coat	9
Single wood floor on joists with metal ceiling	13
Double wood floor on joists	8
Double wood floor on joists with lath and plaster ceiling	5
Single wood floor on joists	14
10-in. reinforced concrete	16
8-in. reinforced concrete	18
6-in. reinforced concrete	25
4-in. reinforced concrete	35
4-in. reinforced concrete with 1-in. wood surface	18

TABLE 12—FLOORS LAID ON GROUND

Heat loss in B.tu per hour per sq. ft. for ground temperature of 50 F and room temperature of 70 F.

4-in. concrete on ground	7
4-in. concrete on cinder fill	6
4-in. concrete on ground with 1-in. tile surface	7
Single wood floor on concrete	3
Single wood floor on cinder fill	3
Single wood floor on joists on ground	3

York City, 6 F below zero; Cleveland, Ohio, 17 F below zero. Thus it will be seen that the assumed temperature of zero degrees is correct for the central portion of the United States. In the extreme north, lower outside temperature must be assumed in accordance with the Weather Bureau record, while in the extreme south, higher temperatures would be correct. When the difference between the outside and inside temperature is not 70 F, the values given in Tables 3 to 10 must be multiplied by a correction factor. The correction factor is obtained as follows:

Take a fraction whose numerator is the difference between the assumed outside temperature and the required inside temperature and whose denominator is 70 degrees.

At Madison, Wisconsin, an outside temperature of 15 F below zero should be assumed. Then for an inside temperature of 70 F, the value in the table must be multiplied by 85/70 or 17/14. If we want to warm a bath room at Madison to 85 F, the difference between inside and outside temperature is 100 F. Then we must use a correction factor of 100/70 or 10/7.

With these data available on heat losses by transmission from the room or building, and with the previously explained method of computing the heat required to heat up the incoming air, then the total heat requirements can

be computed as given in the following typical example.

The heat that is required is made up of three parts: (a) The heat lost through the walls, doors, windows, floors, ceilings and roofs; (b) heat to warm incoming air; and (c) heat to bring the room and contents up to temperature if the heating is not continuous. If the warming up period is long, less heat is required per hour than if it is short.

Problem: How much heat must be supplied per hour to heat a room continuously if one exposed side is south and another west. The ceiling is ten feet high. The south side is 16 ft. long and has two windows of 15 sq. ft. each. The west side is 20 ft. long, and has one window of 15 sq. ft., and one door 3 ft. by 7 ft. The outside walls are 9-in. brick, plastered. The north wall is next to a heated room. The east wall is next to an unheated room. The inside walls are made up with studs, lathed and plastered on both sides. The floor is double wood flooring over an unheated basement. The ceiling is lath and plastered with a finish plaster coat, this built on joists with single wood floor above. There are two changes of air required per hour. The required inside temperature is 75 F and the average expected minimum outside temperature is 5 F below zero (Temperature difference 80 F.)

Computations

South Wall:

Area = $(16 \times 10) - (2 \times 15) = 130$ sq. ft.

Transmission constant for 70 F temperature difference = 22 (Table 3).

For 80 F temperature difference this constant will be

$$\frac{80}{70} \times 22.$$

Heat loss will, therefore, be

$$\frac{80}{70} \times 22 \times 130 = \textbf{3,269 B.tu per hr.}$$

South Wall Windows:

Area of 2 windows = 30 square feet.

Transmission constant for 70 F temperature difference = 78 (Table 8).

For 80 F temperature difference this constant will be

$$\frac{80}{70} \times 78.$$

Heat loss will be, therefore,

$$\frac{80}{70} \times 78 \times 30 = \textbf{2,674 B.tu per hr.}$$

West Wall:

Area = $(20 \times 10) - 15 - (3 \times 7) = 164$ sq. ft.

$$\text{Transmission constant} = \frac{80}{70} \times 22.$$

$$\text{Heat loss will be, therefore, } 164 \times \frac{80}{70} \times 22 + 10 \text{ per cent of } \left(164 \times \frac{80}{70} \times 22\right) = \textbf{4,535 B.tu per hr.}$$

West Wall Window and Door:

Area of window and door = $15 + 21 = 36$ sq. ft.

$$\text{Transmission constant} = \frac{80}{70} \times 78.$$

$$\text{Heat loss} = 36 \times \frac{80}{70} \times 78 + 10 \text{ per cent of } \left(36 \times \frac{80}{70} \times 78\right) = \textbf{3,530 B.tu per hr.}$$

North Wall:

No heat loss on this side as it is next to a heated room.

East Wall (Studs with plaster on both sides):

This wall is next to an unheated room.

Area of wall = $16 \times 10 = 160$ sq. ft.

Transmission constant for lath and plaster wall on studs is 12 (Table 10).

Heat loss, therefore, will be

$16 \times 10 \times 12 =$ **1,920 B.tu per hr.**

Floor (Double floor over an unheated basement):

Area of floor = $16 \times 20 = 320$ sq. ft.

Transmission constant = 8.

Heat loss, therefore, will be

$16 \times 20 \times 8 =$ **2,560 B.tu per hr.**

Ceiling (Lath and plaster with single wood floor above):

Area of ceiling = $16 \times 20 = 320$ sq. ft.

Transmission constant = 9.

Heat loss = $16 \times 20 \times 9 =$ **2,880 B.tu per hr.**

Heat required for two air changes:

Volume of room = $16 \times 20 \times 10 = 3,200$ cu. ft.

Total to be heated 80 F = $2 \times 3,200$ or 6,400 cu. ft.

1 B.tu will heat one cu. ft. of air 50 F.

Heat required to heat up 6,400 cu. ft. 80 F =

$$6400 \times \frac{80}{50} =$$ **10,240 B.tu per hr.**

Total loss **31,608 B.tu per hr.**

Answer

Supplementary Heating

If there is any source of heat other than that to be supplied by the heating system, an estimate of the amount should be made and subtracted from the requirement total. When such allowance is made, consideration must be given as to the continuity and permanence of such supplementary source. Table 13 gives a tabulation of determined values that are available. Other sources may be conservatively estimated in determining the net heating needs for any room.

TABLE 13—HEAT GIVEN UP FROM AUXILIARY SOURCES

OBJECT	B.TU EMITTED PER HOUR
Electric Lamps	Total watts x 3.412
Average person sitting at rest	350
Average person working	450
Cottage cook stove	20,000 to 35,000
Range Boilers	2,000 to 4,000

Heat Emission Constants for Heating Surfaces

The amount of heat emitted from a metallic heating surface per square foot per hour is dependent upon (1) the material of the heating surface, (2) the temperature difference between the heating surface and the air or medium into which the heat is passing, (3) the

velocity of the air or medium passing over the heating surface, (4) the area and position of the exposed heating surface, (5) the nature of the heating medium, (6) the humidity of the air and (7) the thickness of the heating surface metal.

With so many conditions upon which the heat emission is dependent, it is obvious that any constants given must be designated for specific conditions. Table 14 gives the average rate of heat emission from steam heated plates or pipes when the room temperature is approximately 70 F and the steam pressure is about 2 lb. gage. The values given are for transmission into still air. Any draft or forced movement of air over the surface of the pipes or surfaces will increase the heat emission.

TABLE 14

MATERIAL OF SURFACE	HEAT EMITTED PER HOUR PER DEGREE DIFFERENCE OF TEMPERATURE OF SURFACE INTO STILL AIR.
Cast iron—oxidized bare	2
Cast iron—bronzed	2
Cast iron—polished	2 2
Cast iron—painted white or buff	2.2
Steel pipe—Clean—1¼-in. diameter	2.5
Wrought iron pipe—clean—1¼-in. diameter	2.5

The typical steam radiator, as designed, will give apparent results somewhat lower than the values tabulated. This is caused by the travel of the heated air from the bottom of the radiator over the radiating surfaces of the upper part of the radiator.

The intermediate sections necessarily will emit less heat than those at the ends. The end sections will give off more heat per square foot than the amount given in the table on account of the rapid air movement over the surface brought about by the natural air circulation.

Transmission Constants for Insulated Surfaces

The percentage of heat retained by the application of insulation is dependent upon the (1) character of insulation, (2) thoroughness with which the insulation is applied, (3) the thickness of insulation, and (4) protection of insulation from atmospheric changes. Such heat transfer constants as have been determined are usually based upon well-constructed and protected surfaces. Unfortunately, insulation is often maintained in a relatively poor state of repair, and the actual percentage of heat retained is much below that given in the tables in such cases. These tabulated values must be considered, therefore, as the ideal toward which we might approach, but only under very good installation conditions will they be equalled.

The approximate efficiencies for commercial pipe coverings range from about 81 to 88 per cent. These efficiencies are referred to the loss from bare pipe and mean that if the covering is installed correctly, the per cent of heat will be saved that is now being lost from bare pipe as indicated by the efficiencies.

Thickness of Covering to Be Used

For hot water heating and for steam pressures up to ten pounds pressure, one inch commercial pipe covering is usually sufficient. For high pressure steam work one inch covering is sufficient on the smaller pipe sizes, but with piping over three inches in diameter it is good economy to use at least one and one-half inch covering. These thicknesses are based on coal at $6.00.

Editorial Note: An article entitled "Calculating Radiation Requirements" will appear soon. It will show practical applications of the data provided in this article.

"Open for Discussion"

A department in which we follow the custom of technical societies of allotting space in their programs for discussion of the papers presented

STANDARDS FOR AIR CONDITIONING

"Air Conditioning and Human Welfare," *by Major S. Munson Corbett.* Page 42. January, 1930

TO THE sanitary engineer and public health worker nothing will be more satisfactory than a settlement of the perplexing questions which surround the matter of ventilation. At the present time the most encouraging tendency being manifested is the substitution of careful research for personal opinion. The results of this are striking already, and there is no reason to doubt that the future will bring about further advances. We have seen the abandonment of the old idea that organic poisons resulting from respiration constituted the ventilation problem and the replacement of this fallacy by recognition of the importance of the dry bulb temperature. Further refinements were found possible by consideration of wet bulb temperatures and relative humidity of occupied rooms. Accompanying these strides the ventilating engineers have been able to devise methods of removing dust, obnoxious gases, etc., from air entering rooms. The final advance is the establishment of the comfort zone and comfort line by the *A. S. H. V. E.* laboratory.

Major Corbett has brought out an important point which may lead to the next advance in the science of air conditioning. Examination of the results of the *New York State Commission on Ventilation* leads the sanitarian to believe that making comfort the criterion, important as it is, does not provide a complete solution. Other data obtained from schools indicate, although it is possibly not more than an indication, that some other factor not eliminated by a feeling of discomfort, affects the health of children occupying school rooms. While I realize that this conclusion, based upon the much discussed findings mentioned above, will be disagreed with by some, I believe the sanitarian is justified in making so

mild a statement at least. What the health factor may be has not been explained. Possibly low relative humidities with high temperatures may cause certain changes, harmful or otherwise, in the metabolism of the human body. The greater tendency to dryness of the nose and throat surfaces with its effects upon the incidence of respiratory disorders should also be investigated. Perhaps the final result will be a further reduction of the comfort zone or comfort line to a health-comfort zone or line.

In this connection it occurs to me that persons accustomed to higher temperatures the year around, as in the south and southwest, may find that the lower temperatures on the comfort line are not comfortable. In Texas, for instance, I have found an indoor 68 F dry bulb temperature with humidities well within the comfort zone, distinctly uncomfortable. Since, as Major Corbett points out, sufferers from certain respiratory affections frequently find relief in the Southwest, a study of prevailing temperatures and humidities in this section might be of value to physiological research as applied to ventilation.

The sooner medical men recognize the need for study of this obscure relationship which may, and probably does, exist, and show willingness to co-operate with the research worker in ventilation, the sooner will the question be answered. Major Corbett's interest in the matter is an indication of progress. In the meantime the non-medical sanitarian is hopefully waiting.—*E. W. Steel, Professor of Municipal and Sanitary Engineering, A. & M. College of Texas.*

Comment by S. R. Lewis

Other medical men, notably Dr. Hubbard of Toledo, Ohio, have agreed with Dr. Corbett's judgment that the relative humidity should be kept high artificially whenever, due to artificial heat, it becomes extremely low.

Unfortunately, no consistent research, so far as I know, ever has been carried out which demonstrated the effect upon the human body of consistently maintained artificial humidity.

There have been some attempts made to prove or to disprove the theory that the introduction of moisture into school building air should show favorable results in the attendance records, or in the observed cases of respiratory illness.

The New York Commission on Ventilation spent years trying to prove that consistent information about ventilation could be obtained from records of incidence of respiratory diseases in school children. I tried it myself, for a year, moistening the air at one end of a large school in Toledo and leaving dry the other end. All of these tests failed to deliver consistent information.

The New York commission has admitted that no particular constructive information could be deduced about ventilation by health records of public school pupils. It appears that so small a part of the life of each pupil is spent in the public school that we can not expect ever to secure consistent data from public school children.

Observations might be made of the children in some institution such as a blind or deaf and dumb school, where we can control the living conditions of the pupils all the time, not turning them out after a few daylight hours in school into the rain and the snow, to go home and to other enclosed places where widely different conditions of heating and cleanliness of air and of contacts must exist.

Until someone publishes results of authoritative research which prove the benefit of some maintained humidity content, or until the medical profession as a reasonably coherent unit prescribe to the engineering profession something better than the comfort chart of the engineer, the latter, which the engineers developed without the doctors, will rule.

Dr. Corbett raises interesting and possibly vital questions. We should like to have him go further and tell us how we may prove his contentions.

In installations for human ventilation only where the engineers have provided automatic humidity introduction and control, we find a tendency on the part of the owner not to operate the humidifier because of inability to visualize any gain from such operation.

Perhaps some information might be gained by investigating the health conditions of operatives in factories where artificial humidification is maintained. In many plants, due to the necessity for a uniform manufacturing process, the moisture content of the air is controlled. While the operatives are there only perhaps one-third of their time, they may be healthier than the rest of us. I have seen no figures to prove this, however.—*Samuel R. Lewis.*

Multiple Refrigeration Installations

One should never make the mistake of considering a multiple installation as different in fundamental principles from the individual system, for the function of each is the same and the same processes must take place to provide refrigeration whether one unit or several be used in conjunction with one or more refrigerator cabinets.

To insure the success of any installation, one should have a thorough understanding of the underlying principles, i. e., the theory of refrigeration, and of course one must possess a practical working knowledge of the equipment used. The contractor contemplating the installation and servicing of refrigeration equipment will be unwise if he fails to send a capable member of his organization to the factory manufacturing the equipment for a complete course of instruction covering installation and service and he should thoroughly familiarize himself with the construction of the mechanical equipment.

Too much emphasis cannot be placed on the importance of extreme care in all workmanship for an analysis of existing service records discloses that the bulk of all trouble experienced with mechanical refrigeration equipment is a result of some errors in installation and subsequent errors when serviced. The bulk of this is the result of a lack of understanding of the equipment rather than carelessness or a lack of interest on the part of the person doing the work.

One of the greatest causes of trouble with multiple installations is leaks at line connections. Because of the great number of connections in a multiple installation, minute leaks at a number of these will permit the eventual discharge of the refrigerant, and to occasionally add refrigerant without locating these leaks will effect but temporary relief.—*E. Barger.*

Heating·Piping
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American Society of Heating and Ventilating Engineers

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FEBRUARY *1930*

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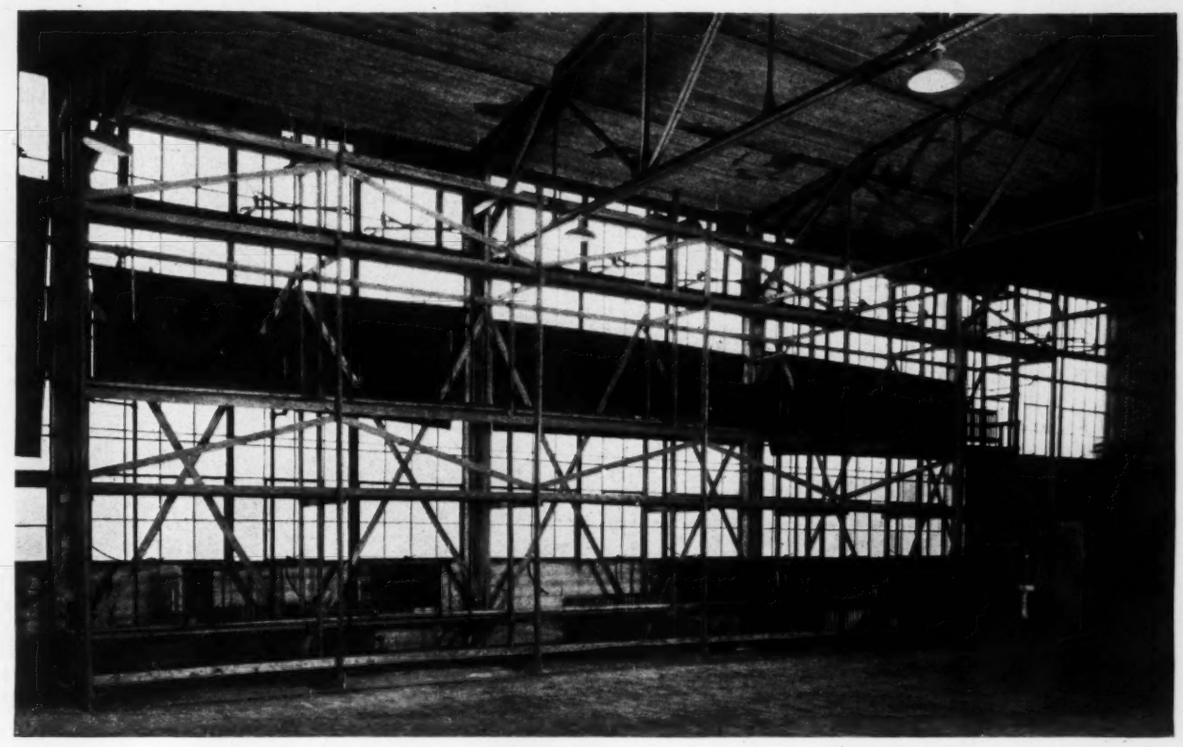

THREE-STORY EXPERIMENTAL INSTALLATION IN THE INTERNAL COMBUSTION ENGINE LABORATORY AT THE AGRICULTURAL AND MECHANICAL COLLEGE OF TEXAS

Pipe and Orifice Sizes for Small Gravity Circulation Hot Water Heating Systems

The results of co-operative research between the A. S. H. V. E. and the Texas Engineering Experiment Station, Dr. F. E. Giesecke, Director

By Elmer G. Smith[1] (*MEMBER*), College Station, Texas

WHEN a hot water heating system is installed in a small building, such as a residence, the two-pipe underfoot feed, direct return system with bare mains is usually chosen. It is cheaper to install than the reversed return system, gives smaller heat losses in the basement, and the piping looks less clumsy. In the past, some trouble has been experienced with the circulation in direct return systems, but recent experiments at this station have demonstrated that when orifices of the proper size are placed in each radiator circuit, the circulation becomes not merely satisfactory but remarkably good. The usual place for such an orifice is in the union at one end of the radiator. There are at least three ways of getting these orifices into the circuit:

1. The contractor may make them out of sheet copper and insert them into the unions as illustrated by Figs. 1 and 2. The copper should be about 0.025 in. thick (22 gage).
2. The contractor may specify valves equipped with orifices of various sizes. There is at least one such valve now on the market. This valve is intended for use with vapor systems, but may also be used with hot water systems.
3. The contractor may specify valves that have adjustable calibrated orifices. There are at least two such valves on the market both of which are intended for use with vapor systems.

In order to make the installations of these systems easy and practical the following tables have been prepared from the results of experimental work:

Table 1, giving the size of riser required to carry a given number of square feet of radiation.

Table 2, giving the size of mains required to carry a

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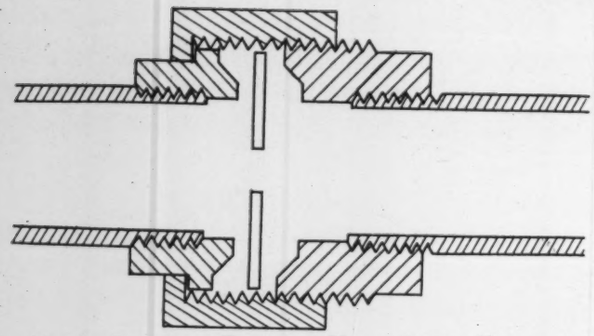

FIG. 1—ORIFICE AS PLACED IN UNION BEFORE UNION IS TIGHTENED

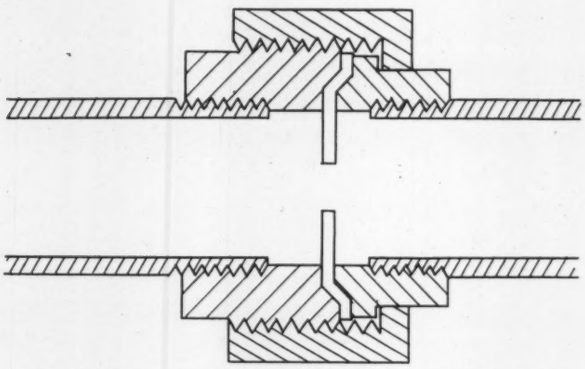

FIG. 2—ORIFICE AS PLACED IN UNION AFTER UNION IS TIGHTENED

given number of square feet of radiation for various lengths of line when the mains are between 3 and 4 ft. above the mid-point of the heater.

Table 3, giving the size of mains required to carry a given number of square feet of radiation when the mains are between 4 and 5 ft. above the mid-point of the heater.

Table 4, giving the size of main heater risers required to carry a given number of square feet of radiation.

Table 5, giving the diameters of the orifices that should be used with various sizes of radiators for different heights above the mid-point of the heater. This table is to be used when the contractor makes his own orifices.

Table 6, giving the sizes of the orifices for use with valves that come equipped with them.

Table 7, giving the proper settings for use with valves with adjustable calibrated orifices.

Experimental Work

The first experiment on a direct return system was made before the tables were calculated. This plant had seven radiators all between 30 sq. ft. and 70 sq. ft. and represented one half of a two-story installation. This system was very successful and the results were fully described in a previous paper entitled "Pipe Sizes for Hot Water Heating Systems," by F. E. Giesecke and the writer which appeared in the June, 1929, JOURNAL of the A. S. H. V. E. After a series of tests on this system and also on a reversed return system, the theory was verified sufficiently to justify an attempt to calculate the tables.

In their present form the tables are not designed to produce the best possible circulation, but represent rather the simplest tables that could be produced and still be assured of reasonably good circulation. When the tables had finally taken shape the system shown in Fig. 11 was designed in accordance with them. When the system was actually built it was necessary to bend the mains back upon themselves in order to get so large a system into the space that was available. Also, the only heater available was a converter intended to heat water for shower baths. The openings in this heater were only 3 in. instead of 3½ in. or 4 in., as called for by the tables. In addition to this the vertical distance from the mid-point of the heater to the mid-point of the mains directly above was made 4 ft., the minimum distance for which Table 3 can be used.

Both mains were bare throughout their entire length and the fact that they were not covered by a tight floor probably tended to cause the radiators near the end of the line to receive cooler water than they otherwise would have.

The results of this test are shown in the table below.

In order to more readily visualize the meaning of these figures, it will be helpful to divide the radiators into two groups (1) medium-sized radiators having more than 35 sq. ft. of surface and (2) small-sized radiators having less than 16 sq. ft. of surface.

RECORD OF TESTS CONDUCTED ON EXPERIMENTAL PLANT BUILT ACCORDING TO TABLES 1, 3, 4, AND 6.

TIME REQUIRED FOR WATER TO REACH LAST RISER, 17 MINUTES

READINGS TAKEN 1 HOUR AFTER STARTING HEATER

Radiator Number	11	12	13	14	15	21	22	23	24	25	31	32	33	34	35
Temperature of Water Entering the Radiator	178	177	172	163	159	173	169	171	164	156	172	163	166	152	145
Temperature of Water Leaving the Radiator	132	140	115	140	132	138	139	141	137	132	141	138	139	129	125
Average Temperature of Water in the Radiator	150	158.5	143.5	152.5	145.5	155.5	154	156	150.5	144	156.5	150.5	152.5	140.5	135

READINGS TAKEN 4½ HOURS AFTER STARTING HEATER

Radiator Number	11	12	13	14	15	21	22	23	24	25	31	32	33	34	35
Temperature of Water Entering the Radiator	200	199	193	189	184	198	191	194	186	177	198	185	192	170	165
Temperature of Water Leaving the Radiator	146	160.5	135	165	148	157.5	159	160	162	155	164	152	158	146	144
Average Temperature of Water in the Radiator	173	179.7	164	177	166	177.7	175	177	174	166	181	168.5	175	158	182

Group 1—Medium-Sized Radiators			Group 2—Small-Sized Radiators		
Radiator Number	Surface in Sq. Ft.	Average Temp. of Water—F	Radiator Number	Surface in Sq. Ft.	Average Temp. of Water—F
11	74¼	173			
12	38½	179.7			
			13	13¾	166
14	74¼	177			
			15	13¾	164
21	74¼	177.7			
22	38½	175			
23	74¼	177			
24	74¼	174			
			25	13¾	166
31	75	181			
32	55	168.5			
33	75	175			
			34	15	163
			35	15	161

It will be noted that the temperatures in the medium-sized radiators are very closely grouped at about 176 F. The hottest is No. 31 (181 F), and the coldest is No. 32 (168.5 F), and the temperature difference between these extremes is only 13½ F, making the deviation from the mean only 6¼ F which is unusually small. These two radiators happen to be next to each other on the second floor and in the half of the line nearer the heater. This seems to indicate that such differences as do exist are caused, not by the position on the line, but rather by the impossibility of selecting exactly the correct resistance for any given radiator, due to the fact that the resistances of both the risers and the orifices vary by jumps.

Referring again to radiators No. 31 and No. 32, it was noticed by reference to Table 6 that the (31-40) orifice called for by radiator No. 31 has a range of 72 sq. ft. to 90.4 sq. ft. Radiator No. 31 is near the low end of this range. Its temperature will therefore be a little high. Similarly, the (21-30) orifice required by radiator No. 32 has a range of 43.9 sq. ft. to 58.5 sq. ft. Radiator No. 32 has 55 sq. ft. of surface. It is therefore near the upper end of this range and its average temperature will therefore be a little low.

Referring to the very small radiators, it is evident that although these radiators agree closely among themselves, they are on the average about 13 F cooler than the larger radiators and that the coldest radiator in this group is a full 20 F cooler than the hottest one in the larger group. This represents a variation of 10 F from the mean. Small radiators dissipate heat more efficiently than larger ones, but even when that is taken into consideration the small radiators are 10 per cent to 15 per cent below the larger ones in the number of B.tu per square foot that they dissipate. As a rule this difference would probably not be noticed by the occupants of the building, but it may be advisable to increase the surface of very small radiators by about 10 per cent to obviate any possibility of having trouble.

How to Select Sizes of Pipes and Orifices

When the various tables are applied in every-day work it will probably be remarked that when mixed pipe sizes are used the flow main or riser, as the case may be, is always smaller than the return main or riser. There are two reasons for this: First, the heat losses due to cooling in the pipes are less with this arrangement than when the flow main or riser is larger; and second, it helps the circulation slightly because cooling in a flow riser lowers the average temperature of a radiator, whereas, cooling in a return riser simply produces more rapid circulation of the water.

In the case of the risers it will also be noticed that the return riser is sometimes two sizes larger than the flow riser as, for instance, a ¾ in. flow riser may be used with a 1¼ in. return riser. This is done to help equalize the friction in the risers.

One-Story Residence Problem

The simplest kind of plant to install is one intended to heat a one-story residence. Such a plant is illustrated in Fig. 3. Before proceeding with the selection of the pipe sizes it may be well to consider some of the piping details. The risers may be connected to the mains as illustrated in Fig. 4A, Fig. 4B or Fig. 4C. Since Fig. 4B is already in common use, it will be selected for this problem. The radiators may be connected with a low flow connection illustrated in Fig. 5A, or with a high flow connection as illustrated in Fig. 5B. The high flow connection locates the valve more conveniently and heats more efficiently. This connection will therefore be selected. In order to be able to refer to them conveniently, the radiators have been numbered consecutively beginning at the heater. The left-hand branch begins with radiator No. 1, and the right-hand branch with radiator No. 8.

How to Select Riser Sizes

By reference to Table 1, it will be noted that the riser sizes should be as follows:

	Flow—In.	Return—In
Radiator No. 1 (35 sq. ft.)	¾	1
Radiator No. 2 (63 sq. ft.)	1	1
Radiator No. 3 (22½ sq. ft.)	½	1
Radiator No. 4 (31½ sq. ft)	¾	¾
Radiator No. 5 (33¾ sq. ft.)	¾	1
Radiator No. 6 (57 sq. ft.)	1	1
Radiator No. 7 (57 sq. ft.)	1	1
Radiator No. 8 (60 sq. ft.)	1	1
Radiator No. 9 (86¼ sq. ft.)	1	1½
Radiator No. 10 (75 sq. ft.)	1	1¼

How to Select Sizes for Mains

There are two tables of pipe sizes for mains: Table 2 is to be used when the distance from the mid-point of the heater to the mid-point of the lowest portion of the mains is between 3 and 4 ft. and Table 3 for use when this distance lies between 4 and 5 ft. The mid-point of a heater is a point midway between the flow connection and the return connection, not a point midway between the top of the heater and the floor, as might be supposed. In

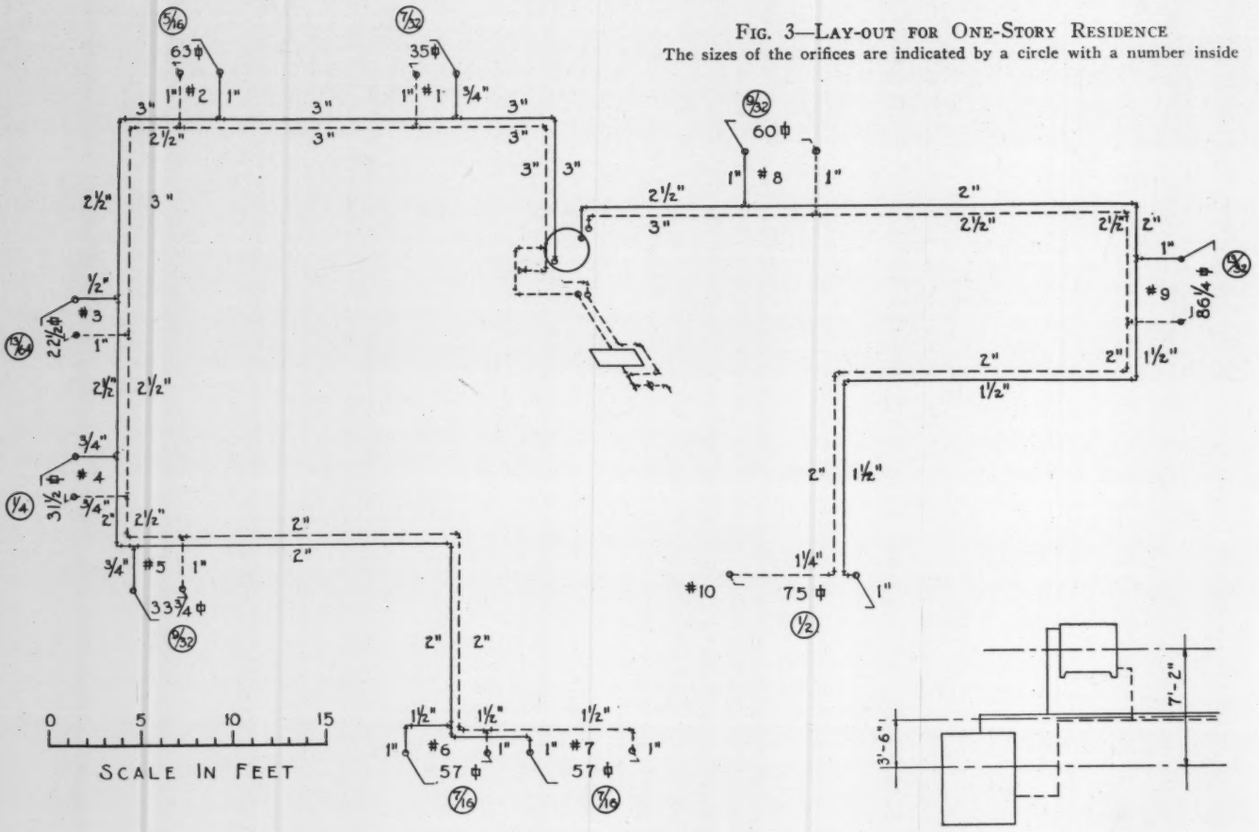

FIG. 3—LAY-OUT FOR ONE-STORY RESIDENCE
The sizes of the orifices are indicated by a circle with a number inside

the case of this particular plant the distance from the mid-point of the heater to the mid-point of the mains is 3 ft. 6 in. Table 2, therefore, must be used.

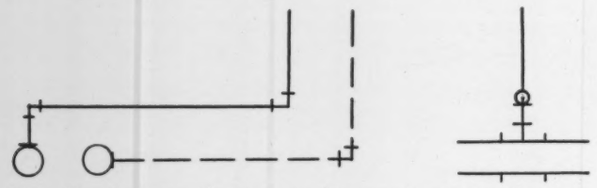

FIG. 4A—LATERAL SECTION OF RISERS (AT LEFT LOOKING ALONG THE MAINS, AT RIGHT LOOKING ACROSS THE MAINS)

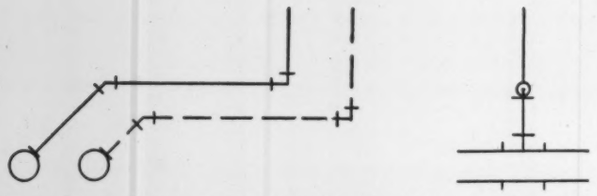

FIG. 4B—LATERAL SECTION OF RISERS (AT LEFT LOOKING ALONG THE MAINS, AT RIGHT LOOKING ACROSS THE MAINS)

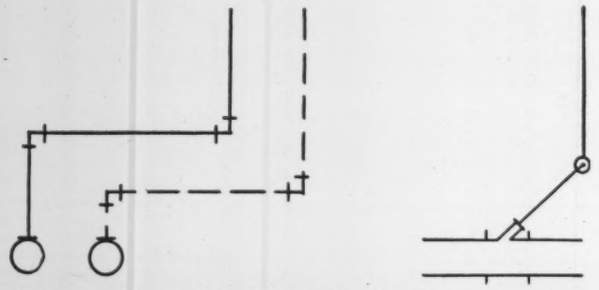

FIG. 4C—LATERAL SECTION OF RISERS (AT LEFT LOOKING ALONG THE MAINS, AT RIGHT LOOKING ACROSS THE MAINS)

In order to select the pipe sizes for a given section of the mains, the length of the line must be known as well as the number of square feet of radiation that the section must carry. The length of a line is the distance, in feet, from the main heater risers to the farthest radiator risers, measured along the mains. In the case of the left-hand branch in Fig. 3, the length of the line is 8 ft. + 23 ft. + 23½ ft. + 18 ft. + 13 ft. + 7 ft. or 92½ ft. Thus, the length of the line lies between 90 ft. and 100 ft.

Accordingly, to find the sizes of mains required between radiator No. 7 and Radiator No. 6, first note

TABLE 1—RISER SIZES*

Pipe Size—In.		Capacity in Sq. Ft.	
Flow	Return		
½	½	0 0	15.4
½	¾	15.4	21.0
½	1	21.0	23.5
¾	¾	23.5	33.2
¾	1	33.2	43.6
¾	1¼	43.6	47.0
1	1	47.0	63.2
1	1¼	63.2	81.0
1	1½	81.0	87.6
1¼	1¼	87.6	130
1¼	1½	130	157
1½	1½	157	196
1½	2	196	251
1½	2½	251	270
2	2	270	384

*All radiators less than 12 ft. above the mid-point of the heater must be connected to the mains by independent risers.

that this section carries 57½ sq. ft. of radiation, and then in the column headed "90-100" find the figures

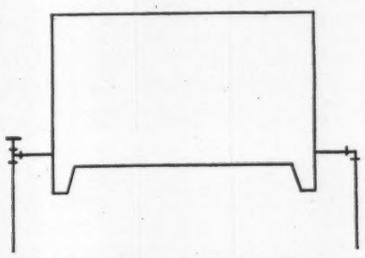

FIG. 5A—LOW FLOW RADIATOR CONNECTION

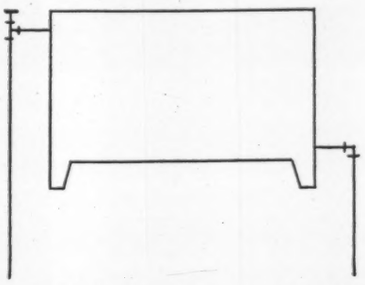

FIG. 5B—HIGH FLOW RADIATOR CONNECTION

49-60. The number of square feet required falls within this range and the correct sizes are found on the same line at the extreme left of the table. Thus, the correct size is 1½ in. for both the flow and return mains.

Proceeding in a similar manner, the remaining sizes are found to be as follows:

SECTION OF LINE BETWEEN RADIATOR	REQUIRED CAPACITY	FLOW—IN.	RETURN—IN.
No. 7 and No. 6	57 sq. ft.	1½	1½
No. 6 and No. 5	114 sq. ft.	2	2
No. 5 and No. 4	147¼ sq. ft.	2	2½
No. 4 and No. 3	179¼ sq. ft.	2½	2½
No. 3 and No. 2	201¾ sq. ft.	2½	3
No. 2 and No. 1	264¾ sq. ft.	3	3
No. 1 and Heater	299¾ sq ft.	3	3

Since the right-hand branch is 66½ ft. long, its pipe sizes are found in the column headed "60-70." These sizes are as follows:

SECTION OF LINE BETWEEN RADIATOR	REQUIRED CAPACITY	FLOW—IN.	RETURN—IN.
No. 10 and No. 9	75 sq. ft.	1½	2
No. 9 and No. 8	161¼ sq. ft.	2	2½
No. 8 and Heater	221¼ sq. ft.	2½	3

How to Select Sizes for the Main Heater Risers

The main risers for the first branch must carry 299¾ sq. ft. of radiation. Therefore, by reference to Table 4 it will be noted that they should be at least 2½ in. The main risers for the right branch must carry 221¼ sq. ft., and again, by reference to Table 4 it will be seen that they, too, should be at least 2½ in.

How to Select Orifice Sizes for Orifices to be Made on the Job

In order to make allowance for the higher pressures that exist close to the heater, it is necessary to use smaller orifices near to the heater and larger ones farther away. To accomplish this, each line is divided into four sections of equal length. The section nearest the heater is No. 1, and the section farthest from the heater is No. 4. Thus, for the left hand branch in Fig. 3, each section is $\frac{92\frac{1}{2}}{4}$ or 23⅛ ft. long.

This distributes the radiators among the sections as follows:

Section No. 1 contains radiator No. 1
Section No. 2 contains radiators No. 2 and No. 3
Section No. 3 contains radiators No. 4 and No. 5
Section No. 4 contains radiators No. 6 and No. 7

In addition to the section in which a radiator is located, it is necessary to know its height above the mid-point of the heater. Since it has been decided to use the high flow connection the distance above the mid-point of the heater must be measured to the mid-point of the radiator. In the system under consideration the radiators are all 26 in. and their mid-points are all 7 ft. 2 in. above the mid-point of the heater. Consequently, in Table 5 only the column headed "7-8" is of interest in this case. The actual selections can now be made.

Radiator No. 1 has a surface of 35 sq. ft. Therefore, in Table 5, Section 1, find the numbers "33.2-37.2" in the column headed "7-8." This is the required range and accordingly the orifice size is found on the same line at the extreme left. In this case it is 7/32 in. For radiators 2 and 3 use Table 5, Section 2, and find, in a manner similar that No. 2 (63 sq. ft.), requires a 5/16 orifice and No. 3 (22½ sq. ft.) requires a 13/64 in. orifice. For radiators 4 and 5 use Table 5, Section 3, and find that radiator 4 (31½ sq. ft.) takes a ¼ in. orifice, and radiator 5 (33¾ sq. ft.) takes a 9/32 in. orifice. Similarly, using Table 5, Section 4, radiators 6 and 7 (each 57 sq. ft.) take 7/16 in. orifices.

For the right-hand branch each section is about 16 2/3 ft. long. Therefore, find radiator 8 in Section 1. Radiator 9 is on the border line between Section 2 and Section 3. In a case like this it is always safer to place the radiator in the section nearest the heater. Therefore, radiator 9 is placed in Section 2. Radiator 10 is in Section 4. Proceeding in the same manner as for the left-hand branch, find that the sizes of the orifices required are as follows:

Radiator 8 (60 sq. ft.) takes a 9/32 in. orifice (Table 5, Section 1).
Radiator 9 (86¼ sq. ft.) takes a 13/32 in. orifice (Table 5, Section 2).
Radiator 10 (75 sq. ft.) takes a ½ in. orifice (Table 5, Section 4).

The design of the system is now complete except for the expansion tank which is of the usual size and installed in the usual manner. The sizes of the orifices are indicated on the plan by a circle with a number inside.

How to Select Orifice Sizes When the Orifices Are Rated in Square Feet of Vapor Radiation

If, instead of making the orifices on the job, it is preferable that valves be purchased already equipped with orifices having their capacity in square feet of vapor radiation marked on them, the procedure is

TABLE 2—CAPACITIES OF MAINS IN SQUARE FEET OF DIRECT RADIATION (DISTANCE FROM MID-POINT OF HEATER TO LOWER PORTIONS OF THE MAINS 3 FT.-4 FT.)

Pipe Sizes		Length of Line in Feet																				Pipe Sizes	
Flow	Return	0	20	20	25	25	30	30	35	35	40	40	45	45	50	50	60	60	70	70	80	Flow	Return
¾	¾	0	23	0	20	0	18	0	17	0	16	0	15	0	15	0	14	0	13	0	12	¾	¾
¾	1	23	30	20	26	18	24	17	22	16	21	15	20	15	20	14	18	13	17	12	15	¾	1
1	1	30	43	26	39	24	36	22	33	21	31	20	29	20	28	18	25	17	23	15	21	1	1
1	1¼	43	57	39	53	36	49	33	45	31	41	29	40	28	38	25	34	23	32	21	34	1	1¼
1¼	1¼	57	83	53	80	49	75	45	70	41	67	40	62	38	59	34	54	32	51	34	47	1¼	1¼
1¼	1½	83	101	80	96	75	89	70	84	67	80	62	75	59	71	54	64	51	61	47	56	1¼	1½
1½	1½	101	130	96	120	89	110	84	105	80	100	75	95	71	90	64	80	61	75	56	70	1½	1½
1½	2	130	169	120	155	110	135	105	136	100	129	95	122	90	117	80	106	75	99	70	92	1½	2
2	2	169	240	155	220	135	205	136	195	129	180	122	170	117	165	106	155	99	145	92	135	2	2
2	2½	240	285	220	264	205	240	195	235	180	220	170	195	165	200	155	187	145	175	135	165	2	2½
2½	2½	285	350	264	330	240	315	235	295	220	280	195	270	200	255	187	235	175	220	165	210	2½	2½
2½	3	350	450	330	442	315	400	295	377	280	360	270	345	255	330	235	300	220	285	210	272	2½	3
3	3	450	630	442	585	400	545	377	520	360	500	345	475	330	460	300	430	285	400	272	385	3	3
3	3½	630	730	585	683	545	640	520	610	500	585	475	558	460	540	430	510	400	470	385	450	3	3½
3½	3½	730	870	683	820	640	780	610	740	585	705	558	680	540	660	510	620	470	580	450	550	3½	3½
3½	4	870	997	820	925	780	880	740	840	705	800	680	770	660	740	620	700	580	670	550	622	3½	4
4	4	997	1170	925	1080	880	1020	840	970	800	930	770	890	740	850	700	800	670	760	622	720	4	4
4	5	.....	.....	1080	1370	1020	1290	970	1230	930	1190	890	1140	850	1100	800	1030	760	980	720	925	4	5
5	5	.....	.....	1370	1860	1290	1770	1230	1700	1190	1650	1140	1600	1100	1540	1030	1460	980	1380	925	1300	5	5
5	6	.....	.....	1860	2250	1770	2170	1700	2080	1650	1960	1600	1970	1540	1900	1460	1800	1380	1700	1300	1600	5	6
6	6	.....	.....	2250	2840	2170	2800	2080	2700	1960	2650	1970	2560	1900	2490	1800	2340	1700	2200	1600	2100	6	6

		80	90	90	100	100	120	120	140	140	160	160	180	180	200	200	240	240	280				
¾	¾	0	11	0	10	0	9	0	8	0	7	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	¾	¾
¾	1	11	14	10	13	9	12	8	10	7	9	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	¾	1
1	1	14	20	13	19	12	17	10	15	9	14	0	13	0	12	0	11	0	10	.....	.....	1	1
1	1¼	20	28	19	26	17	23	15	21	14	20	13	18	12	17	11	16	10	15	.....	.....	1	1¼
1¼	1¼	28	44	26	41	23	38	21	34	20	32	18	30	17	29	16	29	15	27	.....	.....	1¼	1¼
1¼	1½	44	54	41	49	38	46	34	41	32	39	30	36	29	35	29	34	27	31	.....	.....	1¼	1½
1½	1½	54	65	49	60	46	58	41	52	39	50	36	45	35	45	34	40	31	35	.....	.....	1½	1½
1½	2	65	87	60	81	58	76	52	69	50	67	45	61	45	60	40	53	35	48	.....	.....	1½	2
2	2	87	130	81	125	76	110	69	105	67	100	61	95	60	90	53	80	48	75	.....	.....	2	2
2	2½	130	153	125	150	110	136	105	127	100	118	95	110	90	103	80	93	75	88	.....	.....	2	2½
2½	2½	153	185	150	190	136	180	127	160	118	145	110	130	103	120	93	110	88	105	.....	.....	2½	2½
2½	3	185	247	190	246	180	235	160	214	145	189	130	171	120	160	110	146	105	138	.....	.....	2½	3
3	3	247	370	246	350	235	340	214	320	189	270	171	250	160	240	146	215	138	200	.....	.....	3	3
3	3½	370	434	350	410	340	390	320	367	270	322	250	300	240	286	215	254	200	237	.....	.....	3	3½
3½	3½	434	525	410	500	390	460	367	430	322	400	300	370	286	350	254	310	237	290	.....	.....	3½	3½
3½	4	525	596	500	570	460	524	430	490	400	460	370	438	350	410	310	368	290	344	.....	.....	3½	4
4	4	596	690	570	600	524	610	490	570	460	540	438	510	410	490	368	450	344	420	.....	.....	4	4
4	5	690	890	600	850	610	790	570	738	540	700	510	668	490	644	450	600	420	560	.....	.....	4	5
5	5	890	1260	850	1200	790	1120	738	1050	700	1000	668	970	644	940	600	890	560	850	.....	.....	5	5
5	6	1260	1550	1200	1488	1120	1400	1050	1315	1000	1255	970	1226	940	1160	890	1120	850	990	.....	.....	5	6
6	6	1550	2000	1488	1950	1400	1850	1315	1760	1255	1680	1226	1620	1160	1580	1120	1520	990	1470	.....	.....	6	6

TABLE 3—CAPACITIES OF MAINS IN SQUARE FEET OF DIRECT RADIATION (DISTANCE FROM MID-POINT OF HEATER TO MID-POINT OF MAIN DIRECTLY ABOVE 4 FT.-5 FT.)

Pipe Sizes		Length of Line in Feet																				Pipe Sizes	
Flow	Return	0	20	20	25	25	30	30	35	35	40	40	45	45	50	50	60	60	70	70	80	Flow	Return
¾	¾	0	27	0	23	0	21	0	20	0	18	0	17	0	17	0	16	0	15	0	14	¾	¾
¾	1	27	35	23	30	21	28	20	25	18	24	17	23	17	23	16	21	15	20	14	17	¾	1
1	1	35	50	30	45	28	41	25	38	24	36	23	33	23	32	21	29	20	27	17	24	1	1
1	1¼	50	66	45	61	41	57	38	52	36	47	33	46	32	44	29	39	27	37	24	39	1	1¼
1¼	1¼	66	96	61	92	57	87	52	81	47	77	46	71	44	68	39	62	37	59	39	54	1¼	1¼
1¼	1½	96	117	92	111	87	103	81	97	77	92	71	87	68	82	62	74	59	70	54	65	1¼	1½
1½	1½	117	150	111	139	103	127	97	121	92	115	87	110	82	104	74	92	70	86	65	81	1½	1½
1½	2	150	195	139	179	127	156	121	157	115	149	110	141	104	135	92	122	86	104	81	106	1½	2
2	2	195	278	179	254	156	237	157	226	149	208	141	196	135	191	122	179	104	167	106	156	2	2
2	2½	278	330	254	294	237	278	226	272	208	254	196	225	191	231	179	216	167	202	156	191	2	2½
2½	2½	330	404	294	381	278	364	272	341	254	324	225	312	231	295	216	271	202	254	191	242	2½	2½
2½	3	404	520	381	488	364	462	341	436	324	415	312	398	295	381	271	347	254	329	242	314	2½	3
3	3	520	728	488	676	462	630	436	600	415	577	398	548	381	532	347	497	329	462	314	445	3	3
3	3½	728	842	676	790	630	740	600	705	577	676	548	645	532	624	497	590	462	543	445	520	3	3½
3½	3½	842	1005	790	974	740	900	705	855	676	814	645	785	624	762	590	717	543	670	520	635	3½	3½
3½	4	1005	1150	974	1070	900	1020	855	970	814	925	785	888	762	854	717	808	670	775	635	718	3½	4
4	4	1150	1350	1070	1250	1020	1180	970	1120	925	1075	888	1030	854	981	808	924	775	877	718	832	4	4
4	5	.....	.....	1250	1580	1180	1490	1120	1420	1075	1375	1030	1320	981	1270	924	1190	877	1030	832	1070	4	5
5	5	.....	.....	1580	2150	1490	2050	1420	1960	1375	1901	1320	1850	1270	1780	1190	1690	1030	1600	1070	1500	5	5
5	6	.....	.....	2150	2600	2050	2510	1960	2400	1901	2260	1850	2280	1780	2200	1690	2080	1600	1965	1500	1780	5	6
6	6	.....	.....	2600	3280	2510	3240	2400	3120	2260	3060	2280	2960	2200	2880	2080	2700	1965	2540	1780	2310	6	6

		80	90	90	100	100	120	120	140	140	160	160	180	180	200	200	240	240	280				
¾	¾	0	13	0	11	0	9	0	9	0	8	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	¾	¾
¾	1	13	16	11	15	9	14	9	11	8	10	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	¾	1
1	1	16	23	15	22	14	20	11	17	10	16	0	15	0	14	0	13	0	11	.....	.....	1	1
1	1¼	23	32	22	30	20	26	17	24	16	23	15	21	14	20	13	18	11	17	.....	.....	1	1¼
1¼	1¼	32	51	30	47	26	44	24	39	23	37	21	35	20	33	18	33	17	31	.....	.....	1¼	1¼
1¼	1½	51	62	47	56	44	53	39	47	37	45	35	41	33	40	33	39	31	36	.....	.....	1¼	1½
1½	1½	62	75	56	69	53	67	47	60	45	58	41	52	40	52	39	46	36	40	.....	.....	1½	1½
1½	2	75	100	69	93	67	88	60	80	58	77	52	70	52	69	46	61	40	55	.....	.....	1½	2
2	2	100	150	93	144	88	127	80	121	77	115	70	110	69	104	61	92	55	86	.....	.....	2	2
2	2½	150	177	144	173	127	157	121	147	115	136	110	127	104	119	92	107	86	102	.....	.....	2	2½
2½	2½	177	214	173	219	157	208	147	185	136	167	127	150	119	138	107	127	102	121	.....	.....	2½	2½
2½	3	214	285	219	284	208	272	185	248	167	219	150	197	138	185	127	169	121	159	.....	.....	2½	3
3	3	285	428	284	404	272	392	248	370	219	312	197	288	185	277	169	248	159	231	.....	.....	3	3
3	3½	428	502	404	473	392	450	370	427	312	372	288	347	277	330	248	293	231	274	.....	.....	3	3½
3½	3½	502	607	473	578	450	532	427	497	372	463	347	427	330	404	293	358	274	335	.....	.....	3½	3½
3½	4	607	688	578	658	532	606	497	567	463	532	427	506	404	473	358	425	335	397	.....	.....	3½	4
4	4	688	796	658	762	606	705	567	660	532	624	506	590	473	575	425	520	397	485	.....	.....	4	4
4	5	796	1030	762	982	705	913	660	852	624	810	590	772	575	743	520	693	485	646	.....	.....	4	5
5	5	1030	1455	982	1387	913	1295	852	1210	810	1156	772	1120	743	1088	693	1030	646	981	.....	.....	5	5
5	6	1455	1790	1387	1720	1295	1620	1210	1520	1156	1450	1120	1420	1088	1340	1030	1290	981	1145	.....	.....	5	6
6	6	1790	2310	1720	2250	1620	2140	1520	2035	1450	1940	1420	1870	1340	1825	1290	1760	1145	1700	.....	.....	6	6

exactly the same except that Table 6 is used instead of Table 5. When this is done, the following sizes are obtained:

Radiator 1 (Section 1, 35 sq. ft.) takes a 21-30 orifice.
Radiator 2 (Section 2, 63 sq. ft.) takes a 71-80 orifice.
Radiator 3 (Section 2, 22½ sq. ft.) takes a 21-30 orifice.
Radiator 4 (Section 3, 31½ sq. ft.) takes a 41-50 orifice.
Radiator 5 (Section 3, 33¾ sq. ft.) takes a 51-60 orifice.
Radiator 6 (Section 4, 57 sq. ft.) takes a 141-150 orifice.
Radiator 7 (Section 4, 57 sq. ft.) takes a 141-150 orifice.
Radiator 8 (Section 1, 60 sq. ft.) takes a 41-50 orifice.
Radiator 9 (Section 2, 86¼ sq. ft.) takes a 101-110 orifice.
Radiator 10 (Section 4, 75 sq. ft.) takes a ½ in. orifice.

It will be noticed that a few inch sizes are included with the vapor sizes. This is because a large first-floor radiator may, in some cases, require a larger orifice than is made for vapor systems. These extra large sizes will, of course, have to be made on the job.

How to Determine the Proper Settings for Valves Having an Adjustable Orifice Calibrated in Square Feet of Vapor Radiation

So far as the writer knows, these valves are made only in the ¾ in. size. This means that it will often be necessary to reduce the size of the flow risers in order to connect the valve. This will probably not cause any serious trouble, if the reduction is made fairly close to the valve. It is, of course, especially important to use a full-sized return union elbow. The number of radiator sizes available is also somewhat more limited than in the two previous cases, but the range is large enough so that these valves work all right on some systems. In determining the valve setting, use Table 7 which differs from Tables 5 and 6 in that the number of square feet of radiation is given in the margin and the settings comprise the body of the table.

TABLE 4—CAPACITIES FOR MAIN HEATER RISERS [a]

PIPE SIZE—IN.	CAPACITY IN SQ. FT.
1¼	0– 89
1½	89– 130
2	130– 220
2½	220– 325
3	325– 528
3½	528– 770
4	770–1000
5	1000–1660
6	1660–2480

a The pipe sizes for main heater risers may be larger than indicated in this table, but they must not be smaller.

To determine the setting for radiator 1, locate in Table 7, Section 1, the column headed "7-8" and find on the line with 35 the number "36," which is the required setting. The setting for radiator 2, may be selected from the column headed "7-8" in Table 7, Section 2. The number 63, however, is not located in the margin of the table, so by interpolation, find that the required setting is about 79.

Proceeding in a similar manner, the valve settings may all be determined as follows:

For

Radiator 1 (Section 1, 35 sq. ft.) the setting is 36.
Radiator 2 (Section 2, 63 sq. ft.) the setting is 80.
Radiator 3 (Section 2, 22½ sq. ft.) the setting is 30.
Radiator 4 (Section 3, 31½ sq. ft.) the setting is 55.
Radiator 5 (Section 3, 33¾ sq. ft.) the setting is 62.
Radiator 6 (Section 4, 57 sq. ft.) the setting is 155.
Radiator 7 (Section 4, 57 sq. ft.) the setting is 155.
Radiator 8 (Section 1, 60 sq. ft.) the setting is 58.
Radiator 9 (Section 2, 86¼ sq. ft.) the setting is 114.
Radiator 10 (Section 4, 75 sq. ft.) the setting is 220.[2]

How to Deal with Branched Circuits

In Fig. 6 is illustrated a larger system than the one

[2] If the valve is calibrated only up to 200 this will be near enough.

FIG. 6—LAY-OUT FOR A TWO-STORY BUILDING WITH BRANCH MAINS

The sizes of the orifices are indicated by a circle with a number inside

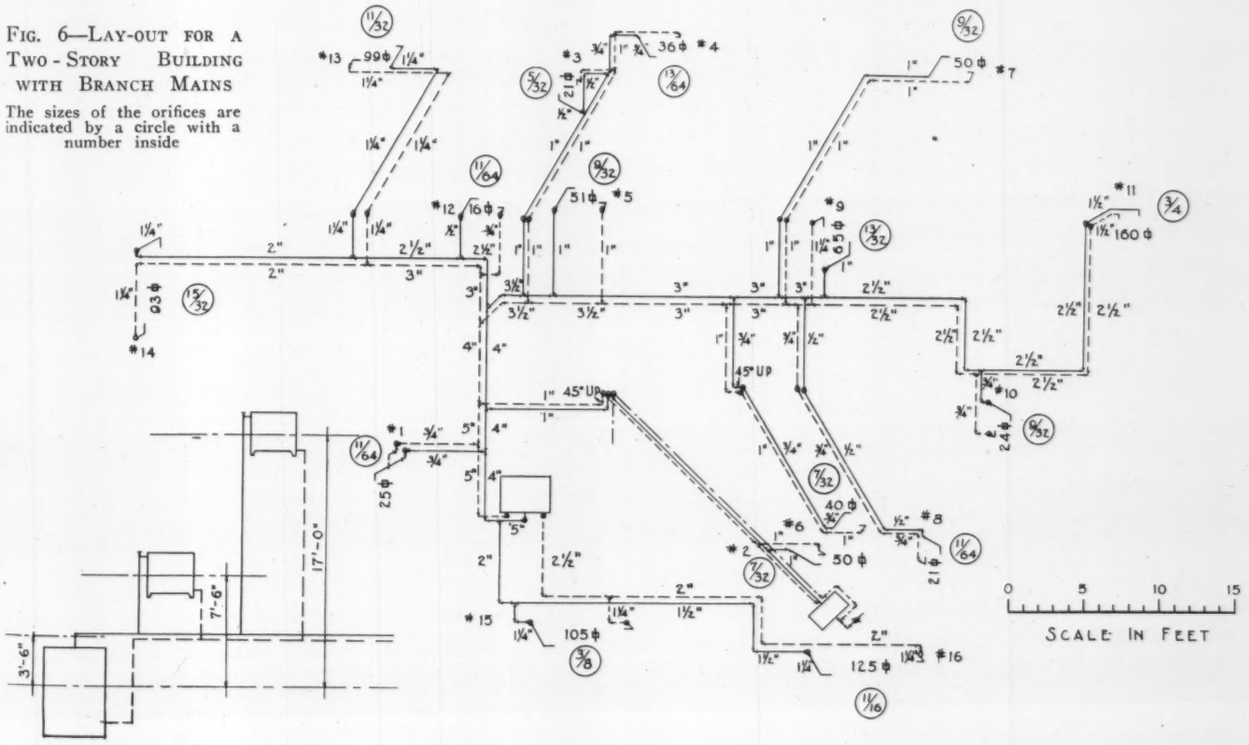

Table 5—Capacities of Orifices in Square Feet of Direct Radiation

Diam. of Orifices in Inches	Section 1 — Height Above Midpoint of Heater in Feet																												Section 1	Diam. of Orifices in Inches
	4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18		
Two																	7.5	10.6	7.8	11.1	8.1	11.5	8.4	11.9	8.7	12.2	8.9	12.6		Two
1/4	6.6	8.6	7.3	9.4	7.8	10.2	8.4	10.9	8.9	11.6	9.4	12.3	9.9	12.9	10.4	13.5	10.6	13.8	11.1	14.3	11.5	14.9	11.9	15.4	12.2	15.9	12.6	16.4		1/4
5/16	8.6	10.8	9.4	11.8	10.2	12.8	10.9	13.8	11.6	14.6	12.3	15.4	12.9	16.1	13.5	16.8	13.8	17.4	14.3	18.1	14.9	18.7	15.4	19.4	15.9	20.6	16.4	20.8		5/16
3/8	10.8	13.4	11.8	14.7	12.8	15.8	13.8	17.0	14.6	18.0	15.4	19.1	16.1	20.0	16.8	21.0	17.4	21.6	18.1	22.5	18.7	23.3	19.4	24.2	20.6	24.9	20.6	25.6		3/8
7/16	13.4	16.1	14.7	17.7	15.8	19.2	17.0	20.4	18.0	21.6	19.1	22.8	20.0	24.2	21.0	25.4	21.6	26.2	22.5	27.1	23.3	28.2	24.2	29.2	24.9	30.1	25.6	30.9		7/16
1/2	16.1	18.7	17.7	20.7	19.2	21.8	20.4	24.5	21.6	26.0	22.8	27.5	24.2	28.8	25.4	30.2	26.2	31.1	27.1	32.4	28.2	33.5	29.2	34.8	30.1	35.8	30.9	36.8		1/2
9/16	18.7	22.0	20.7	24.8	21.8	26.8	24.5	28.8	26.0	30.6	27.5	32.4	28.8	33.8	30.2	35.3	31.1	36.6	32.4	38.2	33.5	39.4	34.8	40.9	35.8	42.1	36.8	43.3		9/16
5/8	22.0	26.0	24.8	28.6	26.8	31.0	28.8	33.2	30.6	35.0	32.4	37.2	33.8	39.0	35.3	40.1	36.6	42.3	38.2	44.1	39.4	45.6	40.9	47.2	42.1	48.6	43.3	50.1		5/8
11/16	26.0	30.0	28.6	33.1	31.0	35.4	33.2	37.2	35.0	40.1	37.2	43.0	39.0	45.0	40.1	47.0	42.3	48.9	44.1	50.9	45.6	52.6	47.2	54.6	48.6	56.2	50.1	57.8		11/16
3/4	30.0	33.6	33.1	37.0	35.4	40.0	37.2	43.0	40.1	46.0	43.0	48.7	45.0	51.1	47.0	53.6	48.9	65.4	50.9	57.7	52.6	59.6	54.6	61.9	56.2	63.7	57.8	65.5		3/4
13/16	33.6	42.9	37.0	47.1	40.0	51.7	43.0	55.5	46.0	59.0	48.7	62.4	51.1	65.3	53.6	68.5	65.4	70.7	57.7	73.6	59.6	76.2	61.9	79	63.7	81.5	65.5	83.6		13/16
7/8	42.9	52.8	47.1	58.1	51.7	63.2	55.5	67.6	59.0	71.9	62.4	76.0	65.3	80.0	68.5	83.0	70.7	86.6	73.6	90.3	76.2	93.4	79	96.8	81.5	99.6	83.6	102		7/8
15/16	52.8	63.0	58.1	69.6	63.2	75.7	67.6	81.5	71.9	85.5	76.0	92.0	80.0	96.5	83.0	101	86.6	105	90.3	109	93.4	113	96.8	117	99.6	120	102	124		15/16
1	63.0	75.1	69.6	81.8	75.7	91.0	81.5	97.5	85.5	104	92.0	110	96.5	115	101	121	105	125	109	130	133	134	117	139	120	143	124	147		1
1 1/16	75.1	86.1	81.8	98.0	91.0	107	97.5	114	104	122	110	128	115	134	121	140	125	146	130	152	134	158	139	163	143	168	147	173		1 1/16
1 1/8	86.1	103	98.0	114	107	124	114	131	122	140	128	148	134	155	140	163	146	170	152	177	158	183	163	190	168	196	173	202		1 1/8
1 3/16	103	117	114	130	124	139	131	149	140	159	148	168	155	172	163	185	170	193	177	201	183	208	190	216	196	222				1 3/16
1 1/4	117	132	130	146	139	159	149	171	159	182	168	193	172	203	185	212	193	222												1 1/4
1 5/16	132	161	146	185	159	201	171	216	182	230	193	244																		1 5/16
1 3/8	161	205	185	228	201	248																								1 3/8

	18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	36	40	40	43	43	46	46	50	50	55	55	60	
Two																													Two
1/4	9.4	13.3	9.9	14.0	10.3	14.5	10.7	15.2	11.1	15.6	11.5	16.2	11.8	17	12.6	17.8	13.3	18.8	13.8	19.5	14.3	20.2	14.9	21.1	15.6	22.1	16.3	23.1	1/4
5/16	13.3	17.3	14.0	18.2	14.5	19	15.2	19.8	15.6	20.3	16.2	21.1	17	22.1	17.8	23.2	18.8	24.5	19.5	25.4	20.2	26.3	21.1	27.4	22.1	28.8	23.1	30	5/16
3/8	17.3	21.8	18.2	22.8	19	23.8	19.8	24.8	20.3	25.5	21.1	26.5	22.1	27.8	23.2	29.0	24.5	30.8	25.4	31.9	26.3	33	27.4	34.4	28.8	36.1	30	37.8	3/8
7/16	21.8	26.1	22.8	28.4	23.8	29.7	24.3	30.9	25.5	31.8	26.5	33	27.8	34.6	29.0	36.2	30.8	38.2	31.9	39.7	33	41.1	34.4	42.9	36.1	45	37.8	47	7/16
1/2	26.1	32.7	28.4	34.4	29.7	35.9	30.9	37.4	31.8	38.4	33	39.8	34.6	41.8	36.2	43.7	38.2	46.2	39.7	48	41.1	49.6	42.9	51.8	45	54.4	47	56.9	1/2
9/16	32.7	39	34.4	40.9	35.9	42.7	37.4	44.5	38.4	45.6	39.8	46.5	41.8	49.7	43.7	52	46.2	55	48	57.2	49.6	59.1	51.8	61.7	54.4	64.6	56.9	67.6	9/16
5/8	39	45.8	40.9	48.2	42.7	50.2	44.5	52.4	45.6	53.8	46.5	55.9	49.7	58.5	52	61.2	55	64.8	57.2	67.2	59.1	69.5	61.7	72.6	64.6	76.1	67.6	79.6	5/8
11/16	45.8	53	48.2	55.6	50.2	58.1	52.4	60.5	53.8	62.1	55.9	64.5	58.5	67.6	61.2	70.8	64.8	74.9	67.2	77.6	69.5	80.4	72.6	83.9	76.1	88	79.6	92	11/16
3/4	53	61.2	55.6	64.3	58.1	67.1	60.5	69.9	62.1	71.8	64.5	74.5	67.6	78.1	70.8	81.8	74.9	86.5	77.6	89.8	80.4	92.9	83.9	96.9	88	102	92	106	3/4
13/16	61.2	69.4	64.3	73.9	67.1	76	69.9	79.2	71.8	81.4	74.5	84.5	78.1	88.6	81.8	92.7	86.5	98	89.8	102	92.9	105	96.9	110	102	115	106	120	13/16
7/8	69.4	88.5	73.9	93	76	97.1	79.2	101	81.4	104	84.5	108	88.6	113	92.7	118	98	125	102	130	105	134	110	140	115	147	120	154	7/8
15/16	88.5	108	93	114	97.1	119	101	124	104	127	108	132	113	139	118	145	125	153	130	159	134	165	140	172	147	180	154	188	15/16
1	108	131	114	138	119	144	124	150	127	154	132	160	139	167	145	175	153	185	159	192	165	199	172	207	180	218	188	227	1
1 1/16	131	156	138	164	144	171	150	178	154	183	160	190	167	199	175	208	185	220	192	229	199								1 1/16
1 1/8	156	183	164	192	171	201	178	209	183	215	190	223	199	234															1 1/8
1 3/16	183	213	192	224																									1 3/16
1 1/4																													1 1/4
1 5/16																													1 5/16
1 3/8																													1 3/8
1 7/16																													1 7/16
1 1/2																													1 1/2
1 9/16																													1 9/16

Diam. of Orifices in Inches	Section 2 — Height Above Midpoint of Heater in Feet																												Section 2	Diam. of Orifices in Inches
	4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18		
Two																	5.9	8.4	6.2	8.7	6.4	9.1	6.6	9.4	6.9	9.7	7.1	10.1		Two
1/4	5.1	6.6	5.6	7.3	6.1	7.9	6.6	8.6	7.0	9.1	7.4	9.7	7.8	10.2	8.2	10.7	8.4	10.9	8.7	11.4	9.1	11.8	9.4	12.2	9.7	12.6	10.1	13.		1/4
5/16	6.6	8.2	7.3	9.1	7.9	10.	8.6	10.8	9.1	11.5	9.7	12.1	10.2	12.8	10.7	13.4	10.9	13.7	11.4	14.3	11.8	14.8	12.2	15.3	12.6	15.8	13.	16.3		5/16
3/8	8.2	10.2	9.1	11.4	10.	12.4	10.8	13.5	11.5	14.2	12.1	15.1	12.8	15.7	13.4	16.5	13.7	17.1	14.3	17.8	14.8	18.5	15.3	19.1	15.8	19.7	16.3	20.4		3/8
7/16	10.2	12.4	11.4	13.8	12.4	15.	13.5	16.	14.2	17.1	15.1	18.1	15.7	19.1	16.5	20.	17.1	20.6	17.8	21.5	18.5	22.3	19.1	23.1	19.7	23.8	20.4	24.6		7/16
1/2	12.4	14.7	13.8	16.1	15.	17.6	16.	19.	17.1	20.4	18.1	21.2	19.1	22.4	20.0	23.8	20.6	24.6	21.5	25.6	22.3	26.6	23.1	27.5	23.8	28.4	24.6	29.3		1/2
9/16	14.7	17.	16.1	19	17.6	20.7	19.0	22.0	20.4	23.5	21.2	25.4	22.4	26.6	23.8	28.	24.6	28.9	25.6	30.1	26.6	31.2	27.5	32.3	28.4	33.4	29.3	34.4		9/16
5/8	17.	19.7	19.	21.4	20.7	23.5	22.	25.9	23.5	27.8	25.4	29.4	26.6	30.9	28.	32.4	28.9	33.4	30.1	34.8	31.2	36.1	32.3	37.3	33.4	38.6	34.4	39.8		5/8
11/16	19.7	22.1	21.4	25.4	23.5	27.7	25.9	29.9	27.8	32.0	29.4	33.6	30.9	35.4	32.4	37.2	33.4	39.3	34.8	40.1	36.1	41.7	37.3	43.1	38.6	44.6	39.8	46.		11/16
3/4	22.1	25.7	25.4	28.7	27.7	31.4	29.9	33.4	32.0	35.7	33.6	38.	35.4	40.	37.2	42.	39.3	43.8	40.1	45.6	41.7	47.3	43.1	48.9	44.6	50.6	46.	52.1		3/4
13/16	25.7	32.8	28.7	36.	31.4	39.4	33.4	42.7	35.7	45.1	38.	48.6	40.	51.4	42.	54.	43.8	55.2	45.6	57.5	47.3	60.4	48.9	62.5	50.6	64.5	52.1	66.6		13/16
7/8	32.8	39.3	36.	44.	39.4	48.7	42.7	52.5	45.1	56.1	48.6	59.5	51.4	62.9	54.	65.5	55.2	68.5	57.5	71.3	60.4	74.	62.5	76.5	64.5	79.	66.6	81.6		7/8
15/16	39.3	46.2	44.	53.5	48.7	58.9	52.5	63.4	56.1	67.4	59.5	71.5	62.9	75.6	65.5	79.2	68.5	82.6	71.3	86.	74.	89.4	76.5	92.5	79.	95.5	81.6	99.4		15/16
1	46.2	57.1	53.5	63.2	58.9	69.2	63.4	75.0	67.4	81.	71.5	83.5	75.6	90.	79.2	95.	82.6	98.4	86.	102.	89.4	106.	92.5	110.	95.5	114.	99.4	117.		1
1 1/16	57.1	66.	63.2	74.1	69.2	81.3	75.0	86.	81.	95.	83.5	100.	90.	106.	95.	111.	98.4	115.	102.	120.	106.	125.	110.	129.	114.	134.	117.	138.		1 1/16
1 1/8	66	77.	74.1	83.6	81.3	95.6	86.	103.	95.	110.	100.	117.	106.	123.	111.	130.	115.	135.	120.	140.	125.	145.	129.	150.	134.	155.	138.	160.		1 1/8
1 3/16	77	84.	83.6	99.0	95.6	108.	103.	117.	110.	125.	117.	131.	123.	138.	130.	145.	135.	153.	140.	159.	145.	165.	150.	171.	155.	177.	160.	182.		1 3/16
1 1/4	84	102.	99.	114.	108.	125.	117.	131.	125.	141.	131.	150.	138.	159.	145.	166.	153.	177.	159.	191.	165.		171.	198.	177.	205.	182.	211.		1 1/4
1 5/16	102	128.	114.	138.	125.	152.	131.	166.	141.	178.	150.	190.	159.	200.	166.	210.	177.	220.	184.	236.	191.		198.	243.						1 5/16
1 3/8	128	151.	138.	172.	152.	189.	166.	204.	178.	220.	190.	234.																		1 3/8
1 7/16	151	184.	172.	208.	189.	228.																								1 7/16
1 1/2	184	219.																												1 1/2
1 9/16																														1 9/16
1 5/8																														1 5/8
1 11/16																														1 11/16
1 3/4																														1 3/4

	18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	33	40	40	43	43	46	46	50	50	55	55	60	
Two																													Two
1/4	7.5	10.6	7.9	11.1	8.2	11.6	8.6	12.2	8.9	12.7	9.3	13.1	9.7	13.8	10.2	14.4	10.7	15.2	11.1	15.8	11.5	16.3	12.	17.	12.6	17.9	13.2	18.7	1/4
5/16	10.6	13.8	11.1	14.4	11.6	15.1	12.2	15.8	12.7	16.5	13.1	17.1	13.8	17.9	14.4	18.7	15.2	19.7	15.8	20.5	16.3	21.2	17.	22.2	17.9	23.2	18.7	24.3	5/16
3/8	13.8	17.3	14.4	18.2	15.1	19.	15.8	19.9	16.5	20.7	17.1	21.4	17.9	22.4	18.7	23.5	19.7	24.8	20.5	25.8	21.2	27.6	22.2	27.8	23.2	29.2	24.3	30.5	3/8
7/16	17.3	21.6	18.2	22.6	19.0	23.6	19.9	24.8	20.7	25.8	21.4	26.6	22.4	28.	23.5	29.2	24.8	30.8	25.8	32.1	27.6	33.2	27.8	34.6	29.2	36.4	30.5	38.	7/16
1/2	21.6	26.1	22.6	27.3	23.6	28.6	24.8	30.	25.8	31.2	26.6	32.2	28.	33.8	29.2	35.4	30.8	37.3	32.1	38.8	33.2	40.1	34.6	41.9	36.4	44.	38.	46.	1/2
9/16	26.1	31.	27.3	32.5	28.6	34.	30.	35.6	31.2	37.1	32.2	38.4	33.8	40.2	35.4	42.1	37.3	44.4	38.8	46.1	40.1	47.7	41.9	49.9	44.	52.4	46.	54.6	9/16
5/8	31.	36.5	32.5	38.2	34.	40.	35.6	42.	37.1	43.6	38.4	45.1	40.2	47.4	42.1	49.5	44.4	52.5	46.1	54.4	47.7	56.1	49.9	58.7	52.4	61.5	54.6	64.4	5/8
11/16	36.5	42.2	38.2	44.2	40.	46.3	42.	48.5	43.6	50.4	45.1	52.1	47.4	54.6	49.5	57.2	52.2	60.4	54.4	62.6	56.1	65.	58.7	67.8	61.5	71.1	64.4	74.4	11/16
3/4	42.2	48.7	44.2	51.1	46.3	53.5	48.5	56.	50.4	58.2	52.1	60.2	54.6	63.2	57.2	66.1	60.4	69.6	62.6	72.5	65.	75.	67.8	78.4	71.1	82.1	74.4	86.	3/4
13/16	48.7	55.3	51.1	57.9	53.5	60.6	56.	63.5	58.2	66.	60.2	68.3	63.2	71.5	66.1	75.	69.6	79.	72.5	82.1	75.	85.	78.4	88.8	82.1	93.1	86.	97.5	13/16
7/8	55.3	70.5	57.9	74.	60.6	77.4	63.5	81.	66.	84.3	68.3	87.1	71.5	91.5	75.	95.8	79.	101.	82.1	105.	85.	109.	88.8	113.	93.1	119.	97.5	124.	7/8
15/16	70.5	86.5	74.	90.6	77.4	94.8	81.	99.4	84.3	103.	87.1	107.	91.5	112.	95.8	117.	101.	124.	105.	129.	109.	133.	113.	139.	119.	146.	124.	152.	15/16
1	86.5	104.	90.6	109.	94.8	114.	99.4	120.	103.	125.	107.	129.	112.	135.	117.	142.	124.	149.	129.	155.	133.	160.	139.	168.	146.	176.	152.	184.	1
1 1/16	104.	124.	109.	130.	114.	136.	120.	143.	125.	148.	129.	154.	135.	161.	142.	169.	149.	178.	155.	185.	160.	191.	168.	200.	176.	210.	184.	219.	1 1/16
1 1/8	124.	146.	130.	153.	136.	160.	143.	168.	148.	174.	154.	180.	161.	189.	169.	198.	178.	208.	185.	217.	191.	224.							1 1/8
1 3/16	146.	170.	153.	178.	160.	186.	168.	195.	174.	203.	180.	210.	189.	220.	198.	230.													1 3/16
1 1/4	170.	193.	178.	202.	186.	212.	195.	222.																					1 1/4
1 5/16	193.	223.																											1 5/16
1 3/8																													1 3/8
1 7/16																													1 7/16
1 1/2																													1 1/2
1 9/16																													1 9/16
1 5/8																													1 5/8
1 11/16																													1 11/16
1 3/4																													1 3/4

TABLE 5—CAPACITIES OF ORIFICES IN SQUARE FEET OF DIRECT RADIATION—*Continued*

Diam. of Orifices in Inches	Section 3 — Height Above Midpoint of Heaters in Feet																										Section 3	Diam. of Orifices in Inches
	4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18	
Two																													Two
1/4																													1/4
3/8	3.6	4.8	4.1	5.3	4.5	5.9	4.9	6.4	5.3	6.8	5.6	7.3	5.9	7.8	6.2	8.1	6.3	8.2	6.6	8.5	6.8	8.9	7.2	9.2	7.4	9.6	7.6	9.9	3/8
1/2	4.8	6.0	5.3	6.7	5.9	7.4	6.4	8.0	6.8	8.6	7.3	9.2	7.8	9.7	8.1	10.1	8.2	10.3	8.5	10.7	8.9	11.2	9.2	11.6	9.6	12.0	9.9	12.4	1/2
5/8	6.0	7.4	6.7	8.4	7.4	9.2	8.0	10.0	8.6	10.7	9.2	11.4	9.7	12.0	10.1	12.6	10.3	12.8	10.7	13.4	11.2	13.9	11.6	14.4	12.0	15.0	12.4	15.4	5/8
3/4	7.4	9.0	8.4	10.1	9.2	11.1	10.0	12.0	10.7	12.9	11.4	13.7	12.0	14.5	12.6	15.2	12.8	15.5	13.4	16.2	13.9	16.8	14.4	17.5	15.0	18.1	15.4	18.6	3/4
7/8	9.0	10.7	10.1	12.0	11.1	13.2	12.0	14.4	12.9	15.4	13.7	16.1	14.5	17.1	15.2	18.0	15.5	18.4	16.2	19.2	16.8	20.0	17.5	20.8	18.1	21.5	18.6	22.2	7/8
1	10.7	12.6	12.0	14.1	13.2	15.6	14.4	16.6	15.4	17.8	16.1	19.0	17.1	20.1	18.0	21.1	18.4	21.7	19.2	22.6	20.0	23.6	20.8	24.5	21.5	25.3	22.2	26.2	1
1 1/4	12.6	14.5	14.1	15.9	15.6	17.5	16.6	19.1	17.8	20.6	19.0	21.4	20.1	22.6	21.1	24.4	21.7	25.1	22.6	26.2	23.6	27.2	24.5	28.2	25.3	29.2	26.2	30.2	1 1/4
1 1/2	14.5	16.1	15.9	18.3	17.5	20.3	19.1	21.4	20.6	23.1	21.4	25.4	22.6	26.8	24.4	28.2	25.1	28.9	26.2	30.2	27.2	31.4	28.2	32.6	29.2	33.8	30.2	34.8	1 1/2
1 3/4	16.1	18.3	18.3	20.8	20.3	22.2	21.4	25.1	23.1	27.0	25.4	28.8	26.8	30.4	28.2	32.0	28.9	32.8	30.2	34.2	31.4	35.7	32.6	37.0	33.8	38.3	34.8	39.5	1 3/4
2	18.3	22.0	20.8	26.6	22.2	29.4	25.1	32.0	27.0	33.1	28.8	36.0	30.4	38.2	32.0	40.2	32.8	41.8	34.2	43.7	35.7	45.5	37.0	47.2	38.3	48.9	39.5	50.5	2
2 1/2	22.0	29.0	26.6	32.9	29.4	35.4	32.0	38.6	33.1	41.8	36.0	43.9	38.2	46.6	40.2	50.3	41.8	51.3	43.7	53.6	45.5	55.7	47.2	57.9	48.9	59.9	50.5	61.9	2 1/2
3	29.0	33.2	32.9	38.2	35.4	42.6	38.6	42.2	41.8	50.7	43.9	54.1	46.6	57.4	50.3	60.4	51.3	61.9	53.6	64.6	55.7	67.3	57.9	70.0	59.9	72.4	61.9	74.6	3
3 1/2	33.2	37.6	38.2	43.6	42.6	51.0	42.2	55.6	50.7	60.0	54.1	63.2	57.4	67.0	60.4	70.5	61.9	73.7	64.6	77.0	67.3	80.2	70.0	83.2	72.4	86.1	74.6	88.9	3 1/2
4	37.6	47.3	43.6	54.4	51.0	60.0	55.6	64.4	60.0	69.5	63.2	74.2	67.0	78.7	70.5	81.0	73.7	86.5	77.0	90.4	80.2	94.0	83.2	97.6	86.1	101.	88.9	104.	4
4 1/2	47.3	54.6	54.4	62.4	60.0	68.1	64.4	74.4	69.5	80.4	74.2	83.4	78.7	91.8	81.0	97.4	86.5	101.	90.4	105.	94.0	110.	97.6	114.	101.	118.	104.	122.	4 1/2
5	54.6	62.4	62.4	69.6	68.1	77.6	74.4	81.5	80.4	92.7	83.4	99.0	91.8	105.	97.4	110.	101.	114.	105.	119.	110.	124.	114.	129.	118.	134.	122.	138.	5
5 1/2	62.4	69.5	69.6	80.0	77.6	84.8	81.5	99.0	92.7	106.	99.0	116.	105.	120.	110.	126.	114.	131.	119.	137.	124.	142.	129.	148.	134.	153.	138.	158.	5 1/2
6	69.5	90.1	80.0	104.	84.8	114.	99.0	125.	106.	129.	116.	138.	120.	144.	126.	155.	131.	165.	137.	173.	142.	180.	148.	187.	153.	194.	158.	200.	6
7	90.1	111.	104.	128.	114.	134.	125.	147.	129.	160.	138.	172.	144.	183.	155.	192.	165.	205.	173.	214.	180.	222.	187.	231.	194.	239.	.....	.....	7
8	111.	134.	128.	143.	134.	162.	147.	178.	160.	195.	172.	207.	183.	226.	192.	232.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	8
9	134.	145.	143.	172.	162.	194.	178.	211.	195.	233.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	9
10	145.	148.	172.	234.	194.	263.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	10
12	148.	201.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	12

	18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	36	40	40	43	43	46	46	50	50	55	55	60	
Two									6.8	9.7	7.1	10.0	7.5	10.6	7.8	11.0	8.3	11.6	8.6	12.1	8.9	12.6	9.3	13.2	9.8	13.8	10.2	14.5	Two
1/4									9.7	12.6	10.0	13.0	10.6	13.7	11.0	14.4	11.6	15.2	12.1	15.8	12.6	16.4	13.2	17.1	13.8	18.0	14.5	18.8	1/4
3/8	8.0	10.5	8.5	11.0	8.9	11.6	9.3	12.1	12.6	15.8	13.0	16.4	13.7	17.2	14.4	18.0	15.2	19.1	15.8	19.8	16.4	20.6	17.1	21.5	18.0	22.6	18.8	23.6	3/8
1/2	10.5	13.1	11.0	13.9	11.6	14.5	12.1	15.2	15.8	19.6	16.4	20.4	17.2	21.5	18.0	22.5	19.1	23.8	19.8	24.7	20.6	25.6	21.5	26.7	22.6	28.1	23.6	29.4	1/2
5/8	13.1	16.4	13.9	17.3	14.5	18.1	15.2	18.9	19.6	23.8	20.4	24.6	21.5	26.0	22.5	27.2	23.8	28.8	24.7	29.9	25.6	31.0	26.7	32.3	28.1	33.9	29.4	35.5	5/8
3/4	16.4	19.8	17.3	20.9	18.1	21.9	18.9	22.8	23.8	28.2	24.6	29.3	26.0	30.8	27.2	32.3	28.8	34.2	29.9	35.5	31.0	35.8	32.3	38.4	33.9	40.4	35.5	42.3	3/4
7/8	19.8	23.5	20.9	24.8	21.9	26.0	22.8	27.1	28.2	33.3	29.3	34.5	30.8	36.4	32.3	38.0	34.2	40.2	35.5	41.8	35.8	43.4	38.4	45.2	40.4	47.5	42.3	49.7	7/8
1	23.5	27.7	24.8	29.2	26.0	30.6	27.1	32.0	28.2	33.3	29.3	34.5	30.8	36.4	32.3	38.0	34.2	40.2	35.5	41.8	35.8	43.4	38.4	45.2	40.4	47.5	42.3	49.7	1
1 1/4	27.7	32.0	29.2	33.8	30.6	35.4	32.0	37.0	33.3	38.5	34.5	39.9	36.4	42.0	38.0	44.0	40.2	46.5	41.8	48.4	43.4	50.1	45.2	52.3	47.5	54.9	49.7	57.5	1 1/4
1 1/2	32.0	36.9	33.8	39.0	35.4	40.8	37.0	42.6	38.5	44.4	39.9	46.1	42.0	48.5	44.0	50.7	46.5	53.7	48.4	55.8	50.1	57.8	52.3	60.4	54.9	63.4	57.5	66.4	1 1/2
1 3/4	36.9	41.9	39.0	44.2	40.8	46.4	42.6	48.4	44.4	50.4	46.1	52.2	48.5	55.0	50.7	57.6	53.7	61.0	55.8	63.2	57.8	65.6	60.4	68.5	63.4	72.0	66.4	75.3	1 3/4
2	41.9	53.5	44.2	56.5	46.4	59.2	48.4	61.8	50.4	64.3	52.2	66.7	55.0	70.2	57.6	73.5	61.0	77.8	63.2	80.7	65.6	83.7	68.5	87.5	72.0	91.9	75.3	96.1	2
2 1/2	53.5	65.5	56.5	69.1	59.2	72.5	61.8	75.7	64.3	78.8	66.7	81.7	70.2	86.0	73.5	90.0	77.8	95.4	80.7	99.0	83.7	103.	87.5	108.	91.9	113.	96.1	118.	2 1/2
3	65.5	79.1	69.1	83.5	72.5	87.5	75.7	91.4	78.8	95.0	81.7	98.6	86.0	104.	90.0	109.	95.4	115.	99.0	119.	103.	124.	108.	129.	113.	136.	118.	142.	3
3 1/2	79.1	94.2	83.5	99.4	87.5	104.	91.4	109.	95.0	113.	98.6	116.	104.	124.	109.	129.	115.	137.	119.	142.	124.	147.	129.	154.	136.	162.	142.	169.	3 1/2
4	94.2	111.	99.4	117.	104.	122.	109.	128.	113.	133.	116.	138.	124.	145.	129.	152.	137.	161.	142.	167.	147.	173.	154.	181.	162.	190.	169.	199.	4
4 1/2	111.	129.	117.0	136.	122.	142.	128.	149.	133.	155.	138.	160.	145.	169.	152.	177.	161.	187.	167.	194.	173.	202.	181.	210.	190.	221.	199.	231.	4 1/2
5	129.	146.	136.	154.	142.	162.	149.	169.	155.	176.	160.	182.	169.	192.	177.	201.	187.	212.	194.	220.	202.	.....	210.	.....	221.	.....	231.	.....	5
5 1/2	146.	168.	154.	177.	162.	185.	169.	194.	176.	201.	182.	209.	192.	220.	201.	.....	212.	.....	220.	.....	.....	.....	.....	.....	.....	.....	.....	.....	5 1/2
6	168.	212.	177.	224.	185.	234.	194.	245.	201.	.....	209.	.....	220.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	6
7	212.	.....	224.	.....	234.	.....	245.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	7
8	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	8
9	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	9
10	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	10
12	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	12

Diam. of Orifices in Inches	Section 4 — Height Above Midpoint of Heater in Feet																										Section 4	Diam. of Orifices in Inches
	4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18	
1/4	2.5	3.3	2.9	3.7	3.2	4.2	3.5	4.5	3.7	4.9	4.0	5.2	4.2	5.5	4.5	5.8	4.5	5.9	4.6	6.0	4.7	6.1	4.9	6.4	5.1	6.7	5.3	6.9	1/4
3/8	3.3	4.1	3.7	4.7	4.2	5.2	4.5	5.7	4.9	6.1	5.2	6.5	5.5	6.9	5.8	7.3	5.9	7.3	6.0	7.4	6.1	7.7	6.4	8.0	6.7	8.4	6.9	8.7	3/8
1/2	4.1	5.1	4.7	5.9	5.2	6.5	5.7	7.1	6.1	7.7	6.5	8.2	6.9	8.9	7.3	9.1	7.3	9.2	7.4	9.3	7.7	9.7	8.0	10.0	8.4	10.4	8.7	10.8	1/2
5/8	5.1	6.2	5.9	7.1	6.5	7.9	7.1	8.6	7.7	9.2	8.2	9.8	8.9	10.4	9.1	11.0	9.2	11.0	9.3	11.1	9.7	11.6	10.0	12.1	10.4	12.6	10.8	13.0	5/8
3/4	6.2	7.4	7.1	8.4	7.9	9.4	8.6	10.2	9.2	11.0	9.8	11.7	10.4	12.4	11.0	13.1	11.0	13.2	11.1	13.2	11.6	13.8	12.1	14.4	12.6	15.0	13.0	15.5	3/4
7/8	7.4	8.7	8.4	9.9	9.4	11.0	10.2	12.0	11.0	12.9	11.7	13.8	12.4	14.6	13.1	15.4	13.2	15.6	13.2	15.5	13.8	16.3	14.4	16.9	15.0	17.6	15.5	18.2	7/8
1	8.7	10.0	9.9	11.4	11.0	12.7	12.0	13.9	12.9	14.9	13.8	15.2	14.6	16.2	15.4	17.1	15.6	17.3	15.5	17.9	16.3	18.8	16.9	19.6	17.6	20.4	18.2	21.1	1
1 1/4	10.0	11.5	11.4	13.2	12.7	14.7	13.9	15.1	14.9	16.4	15.2	17.6	16.2	18.7	17.1	19.8	17.3	20.0	17.9	20.7	18.8	21.7	19.6	22.6	20.4	23.5	21.1	24.4	1 1/4
1 1/2	11.5	13.1	13.2	15.0	14.7	15.5	15.1	17.1	16.4	18.6	17.6	20.0	18.7	21.2	19.8	21.8	20.0	22.4	20.7	23.5	21.7	24.6	22.6	25.6	23.5	26.7	24.4	27.6	1 1/2
1 3/4	13.1	14.8	15.0	18.2	15.5	19.8	17.1	20.4	18.6	22.4	20.0	25.4	21.2	27.2	21.8	28.7	22.4	28.5	23.5	30.0	24.6	31.4	25.6	32.7	26.7	34.0	27.6	35.2	1 3/4
2	14.8	18.2	18.2	21.4	19.8	22.2	20.4	26.8	22.4	29.0	25.4	31.2	27.2	33.2	28.7	34.0	28.5	35.0	30.0	36.8	31.4	38.4	32.7	40.1	34.0	41.8	35.2	43.2	2
2 1/2	18.2	22.0	21.4	23.2	22.2	29.3	26.8	32.4	29.0	33.6	31.2	36.2	33.2	38.8	34.0	41.1	35.0	42.3	36.8	44.4	38.4	46.5	40.1	48.4	41.8	50.5	43.2	52.1	2 1/2
3	22.0	26.2	23.2	30.8	29.3	32.6	32.4	36.5	33.6	40.0	36.2	43.1	38.8	44.7	41.1	49.6	42.3	50.3	44.4	52.8	46.5	55.3	48.4	57.7	50.5	60.1	52.1	62.0	3
3 1/2	26.2	30.7	30.8	33.2	32.6	38.7	36.5	42.8	40.0	45.0	43.1	51.1	44.7	55.1	49.6	57.3	50.3	59.1	52.8	62.0	55.3	65.0	57.7	67.7	60.1	70.5	62.0	72.9	3 1/2
4	30.7	31.6	33.2	38.7	38.7	44.6	42.8	51.3	45.0	55.7	51.7	60.2	55.1	62.9	57.3	66.6	59.1	68.8	62.0	72.2	65.0	75.6	67.7	78.8	70.5	82.0	72.9	84.8	4
4 1/2	31.6	35.8	38.7	43.8	44.6	52.4	51.3	58.1	55.7	63.4	60.2	66.6	62.9	71.4	66.6	75.6	68.8	78.0	72.2	82.0	75.6	85.8	78.8	89.5	82.0	93.2	84.8	96.4	4 1/2
5	35.8	41.2	43.8	46.1	52.4	60.1	58.1	64.5	63.4	70.8	66.6	76.4	71.4	81.9	75.6	82.3	78.0	89.5	82.0	94.0	85.8	98.2	89.5	102.0	93.2	106.0	96.4	111.0	5
5 1/2	41.2	45.2	46.1	64.0	60.1	72.8	64.5	81.5	70.8	91.8	76.4	98.5	81.9	105.0	82.3	111.0	89.5	113.0	94.0	119.0	98.2	124.0	102.0	129.0	106.0	135.0	111.0	141.0	5 1/2
6	45.2	63.2	64.0	77.6	72.8	79.0	81.5	104.0	91.8	113.0	98.5	123.0	105.0	130.0	111.0	130.0	113.0	139.0	119.0	147.0	124.0	153.0	129.0	160.0	135.0	166.0	141.0	172.0	6
7	63.2	76.2	77.6	95.2	79.0	112.0	104.0	125.0	113.0	136.0	123.0	137.0	130.0	146.0	130.0	157.0	139.0	168.0	147.0	177.0	153.0	185.0	160.0	193.0	166.0	201.0	172.0	208.0	7
8	76.2	97.2	95.2	117.0	112.0	134.0	125.0	133.0	136.0	148.0	137.0	163.0	146.0	176.0	157.0	187.0	168.0	201.0	177.0	211.5	185.0	221.0	193.0	230.0	.....	.....	.....	.....	8
9	97.2	132.0	117.0	131.0	134.0	158.0	133.0	184.0	148.0	201.0	163.0	220.0	176.0	238.0	187.0	254.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	9
10	132.0	152.0	131.0	191.0	158.0	210.0	184.0	240.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	10
12	152.0	191.0	191.0	252.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	12

	18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	36	40	40	43	43	46	46	50	50	55	55	60	
1/4	5.7	7.3	6.0	7.8	6.3	8.2	6.6	8.6	6.9	9.0	7.2	9.3	7.6	9.9	7.9	10.3	8.4	11.0	8.8	11.4	9.1	11.8	9.5	12.6	10.0	13.0	10.5	13.6	1/4
3/8	7.3	9.3	7.8	9.8	8.2	10.3	8.6	10.8	9.0	11.3	9.3	11.7	9.9	12.4	10.3	13.0	11.0	13.8	11.4	14.3	11.8	14.8	12.6	15.5	13.0	16.3	13.6	17.1	3/8
1/2	9.3	11.5	9.8	12.2	10.3	12.8	10.8	13.4	11.3	14.1	11.7	14.6	12.4	15.4	13.0	16.1	13.8	17.1	14.3	17.8	14.8	18.5	15.5	19.3	16.3	20.3	17.1	21.2	1/2
5/8	11.5	13.9	12.2	14.8	12.8	15.5	13.4	16.2	14.1	17.0	14.6	17.6	15.4	18.6	16.1	19.5	17.1	20.7	17.8	21.5	18.5	22.3	19.3	23.4	20.3	24.5	21.2	25.7	5/8
3/4	13.9	16.5	14.8	17.6	15.5	18.5	16.2	19.3	17.0	20.2	17.6	20.9	18.6	22.2	19.5	23.2	20.7	24.6	21.5	25.6	22.3	26.6	23.4	27.8	24.5	29.2	25.7	30.6	3/4
7/8	16.5	19.4	17.6	20.6	18.5	21.7	19.3	22.7	20.2	23.8	20.9	24.7	22.2	26.1	23.2	27.3	24.6	29.0	25.6	30.2	26.6	31.2	27.8	32.7	29.2	34.4	30.6	36.0	7/8
1	19.4	22.5	20.6	23.8	21.7	25.1	22.7	26.2	23.8	27.5	24.7	28.5	26.1	30.1	27.3	31.5	29.0	33.5	30.2	34.8	31.2	36.1	32.7	37.8	34.4	39.7	36.0	41.5	1
1 1/4	22.5	26.0	23.8	27.5	25.1	29.0	26.2	30.3	27.5	31.7	28.5	32.9	30.1	34.8	31.5	36.4	33.5	38.7	34.8	40.2	36.1	41.7	37.8	43.7	39.7	45.9	41.5	48.0	1 1/4
1 1/2	26.0	29.4	27.5	31.2	29.0	32.9	30.3	34.4	31.7	36.0	32.9	37.3	34.8	39.4	36.4	41.3	38.7	43.9	40.2	45.6	41.7	47.3	43.7	49.5	45.9	52.0	48.0	54.4	1 1/2
1 3/4	29.4	37.5	31.2	39.8	32.9	42.0	34.4	43.4	36.0	45.9	37.3	47.6	39.4	50.4	41.3	52.6	43.9	56.0	45.5	58.2	47.3	60.4	49.5	63.1	52.0	66.4	54.4	69.5	1 3/4
2	37.5	46.0	39.8	48.9	42.0	51.4	43.4	53.8	45.9	56.3	47.6	58.4	50.4	61.7	52.6	64.6	56.0	68.7	58.2	71.4	60.4	74.0	63.1	77.4	66.4	81.4	69.5	85.1	2
2 1/2	46.0	55.5	48.9	59.0	51.4	62.9	53.8	64.2	56.3	68.0	58.4	70.5	61.7	74.5	64.6	77.9	68.7	83.0	71.4	86.1	74.0	89.3	77.4	93.5	81.4	98.3	85.1	103.0	2 1/2
3	55.5	66.1	59.0	70.3	62.0	74.0	64.2	79.5	68.0	70.9	70.5	84.1	74.5	88.1	77.9	92.8	83.0	98.9	86.1	103.0	89.3	106.0	93.5	111.0	98.3	117.0	103.0	123.0	3
3 1/2	63.1	77.8	70.3	82.5	74.0	87.0	79.5	90.8	70.9	95.0	84.1	98.6	88.1	104.0	92.8	109.0	98.9	116.0	103.0	120.0	106.0	125.0	111.0	131.0	117.0	137.0	123.0	144.0	3 1/2
4	77.8	90.5	82.5	96.0	87.0	101.0	90.8	106.0	95.0	111.0	98.6	115.0	104.0	121.0	109.0	127.0	116.0	135.0	120.0	140.0	125.0	145.0	131.0	152.0	137.0	160.0	144.0	167.0	4
4 1/2	90.5	103.0	96.0	109.0	101.0	115.0	106.0	120.0	111.0	125.0	115.0	130.0	121.0	138.0	127.0	144.0	135.0	154.0	140.0	159.0	145.0	165.0	152.0	173.0	160.0	181.0	167.0	190.0	4 1/2
5	103.0	118.0	109.0	125.0	115.0	132.0	120.0	138.0	125.0	144.0	130.0	149.0	138.0	158.0	144.0	165.0	154.0	175.0	159.0	182.0	165.0	190.0	173.0	198.0	181.0	208.0	190.0	218.0	5
5 1/2	118.0

Table 6—Capacity in Square Feet of Direct Radiation of Orifices Designed for Vapor Systems

Vapor Rating of Orifice		Section 1										Height Above Midpoint of Heater in Feet																Section 1		Vapor Rating of Orifice	
		4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18		
Two	0-10											7.6	10.9	7.0	11.5	8.4	12.	8.7	12.4	9.0	12.8	9.3	13.3	9.7	13.8	10.	14.2	10.4	14.6	Two	0-10
0	10	7.6	15.2	8.4	16.7	9.0	18.0	9.7	19.4	10.4	20.6	10.9	21.6	11.5	22.6	12.	23.9	12.4	24.6	12.8	25.6	13.3	26.6	13.8	27.6	14.2	28.4	14.6	29.2	0	10
11	20	15.2	22.2	16.7	24.8	18.0	27.0	19.4	28.9	20.6	30.8	21.6	32.5	22.6	33.9	23.9	35.4	24.6	37.	25.6	38.4	26.6	39.8	27.6	41.3	28.4	42.5	29.2	43.7	11	20
21	30	22.2	30.1	24.8	33.2	27.0	35.6	28.9	38.3	30.8	40.7	32.5	43.0	33.9	45.0	35.4	47.5	37.	49.3	38.4	51.2	39.8	53.1	41.3	55.	42.5	56.6	43.7	58.3	21	30
31	40	30.1	37.2	33.2	41.0	35.6	44.2	38.3	48.0	40.7	51.1	43.0	54.0	45.0	56.6	47.5	59.3	49.3	61.5	51.2	64.0	53.1	66.3	55.	68.7	56.6	70.5	58.3	72.8	31	40
41	50	37.2	44.1	41.0	49.4	44.2	53.8	48.0	57.6	51.1	61.4	54.	64.5	56.6	67.9	59.3	71.	61.5	74.0	64.0	76.9	66.3	79.6	68.7	82.5	70.5	85.	72.8	87.6	41	50
51	60	44.1	52.2	49.4	57.5	53.8	62.5	57.6	66.9	61.4	71.0	64.5	75.1	67.9	78.0	71.	82.	74.0	86.1	76.9	89.5	79.6	92.8	82.5	96.	85.	99.	87.6	102.	51	60
61	70	52.2	59.2	57.5	66.5	62.5	71.1	66.9	76.5	71.0	81.0	75.1	85.0	78.0	90.5	82.	94.8	86.1	98.5	89.5	102.5	92.8	106.1	96.	110.	99.	113.	102.	117.	61	70
71	80	59.2	66.6	66.5	73.7	71.1	80.2	76.5	85.0	81.0	92.0	85.	97.2	90.5	102.	94.8	107.	98.5	111.	102.5	115.0	106.1	120.	110.	124.	113.	128.	117.	131.	71	80
81	90	66.6	74.2	73.7	81.0	80.2	87.6	85.0	96.1	92.0	103.	97.2	108.	102.	114.	107.	119.	111.	123.	115.0	128.	120.	133.	124.	138.	128.	142.	131.	146.	81	90
91	100	74.2	81.0	81.0	90.5	87.6	98.3	96.1	105.	103.	112.	108.	119.	114.	125.	119.	130.	123.	136.	128.	141.	133.	146.	138.	151.	142.	156.	146.	160.	91	100
101	110	81.0	89.3	90.5	98.6	98.3	107.	105.	115.	112.	123.	119.	129.	125.	134.	130.	141.	136.	146.	141.	154.	146.	159.	151.	165.	156.	169.	160.	175.	101	110
111	120	89.3	97.0	98.6	107.	107.	116.	115.	125.	123.	131.	129.	139.	134.	146.	141.	153.	146.	160.	154.	167.	159.	173.	165.	179.	169.	184.	175.	190.	111	120
121	130	97.0	104.	107.	115.	116.	125.	125.	132.	131.	141.	139.	149.	146.	157.	153.	164.	160.	172.	167.	179.	173.	186.	179.	192.	184.	198.	190.	204.	121	130
131	140	104.	112.	115.	124.	125.	132.	132.	142.	141.	151.	149.	160.	157.	169.	164.	176.	172.	185.	179.	192.	186.	199.	192.	206.	198.	212.	.....	.....	131	140
141	150	112.	120.	124.	130.	132.	141.	142.	151.	151.	162.	160.	171.	169.	182.	176.	189.	185.	197.	192.	205.	199.	212.	.....	.....	.....	.....	.....	.....	141	150
151	160	120.	127.	130.	137.	141.	150.	151.	161.	162.	172.	171.	182.	182.	192.	189.	200.	197.	210.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	151	160
161	170	127.	131.	137.	146.	150.	159.	161.	171.	172.	182.	182.	193.	192.	202.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	161	170
½	....	131.	161.	146.	184.	159.	201.	171.	216.	182.	230.	193.	244.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	½	....
⅝	....	161.	205.	184.	228.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅝	....

		18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	36	40	40	43	43	46	46	50	50	55	55	60		
Two	0-10	10.9	15.4	11.5	16.2	12.0	16.9	12.4	17.8	12.8	18.1	13.3	18.8	14.	19.7	14.6	20.3	15.4	21.8	16.	22.6	16.6	23.4	17.4	24.4	18.1	25.6	18.8	26.8	.....	.....
0	10	15.4	30.8	16.2	32.2	16.9	33.8	17.8	35.3	18.1	36.2	18.8	37.6	19.7	39.4	20.3	41.3	21.8	43.6	22.6	45.2	23.4	46.9	24.4	48.9	25.6	51.4	26.8	53.6	0	10
11	20	30.8	46.2	32.2	48.5	33.8	50.6	35.3	52.9	36.2	54.2	37.6	56.2	39.4	59.0	41.3	61.7	43.6	65.2	45.2	67.6	46.9	70.1	48.9	73.2	51.4	76.8	53.6	80.1	11	20
21	30	46.2	61.6	48.5	64.6	50.6	67.6	52.9	70.5	54.2	72.4	56.2	75.0	59.0	78.7	61.7	82.4	65.2	87.	67.6	90.2	70.1	93.5	73.2	97.5	76.8	103.	80.1	107.	21	30
31	40	61.6	77.0	64.6	80.7	67.6	84.5	70.5	88.0	72.4	90.4	75.0	93.6	78.7	98.5	82.4	103.	87.	109.	90.2	113.	93.5	117.	97.5	122.	103.	128.	107.	134.	31	40
41	50	77.0	92.5	80.7	97.	84.5	102.	88.	106.	90.4	109.	93.6	113.	98.5	118.	103.	124.	109.	131.	113.	136.	117.	140.	122.	147.	128.	154.	134.	161.	41	50
51	60	92.5	108.	97.	113.	102.	118.	106.	123.	109.	126.	113.	131.	118.	138.	124.	144.	131.	152.	136.	158.	140.	164.	147.	171.	154.	179.	161.	187.	51	60
61	70	108.	123.	113.	129.	118.	135.	123.	141.	126.	145.	131.	150.	138.	158.	144.	165.	152.	174.	158.	181.	164.	187.	171.	195.	179.	203.	187.	214.	61	70
71	80	123.	139.	129.	146.	135.	152.	141.	159.	145.	163.	150.	169.	158.	177.	165.	186.	174.	196.	181.	204.	187.	211.	195.	220.	.....	.....	.....	.....	71	80
81	90	139.	154.	146.	162.	152.	169.	159.	178.	163.	181.	169.	188.	177.	197.	186.	206.	196.	218.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	81	90
91	100	154.	169.	162.	178.	169.	186.	178.	194.	181.	199.	188.	206.	197.	217.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	91	100
101	110	169.	185.	178.	193.	186.	202.	194.	211.	199.	217.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	101	110
111	120	185.	201.	193.	211.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	111	120
121	130	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	121	130
131	140	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	131	140
141	150	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	141	150
151	160	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	151	160
161	170	Two 0-10 means that two 0-10 orifices are to be placed in the radiators, one in each union.																												161	170
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....

Vapor Rating of Orifice		Section 2										Height Above Midpoint of Heater in Feet																Section 2		Vapor Rating of Orifice	
		4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18		
Two	0-10																	6.9	9.7	7.2	10.1	7.4	10.5	7.8	10.9	8.	11.2	8.2	11.6	Two	0-10
0	10	5.8	11.6	6.4	13.0	7.1	14.1	7.6	15.1	8.1	16.1	8.6	17.	9.1	18.	9.5	18.8	9.7	19.4	10.1	20.2	10.5	21.	10.9	21.8	11.2	22.5	11.6	23.2	0	10
11	20	11.6	17.2	13.0	18.6	14.1	20.4	15.1	22.	16.1	24.1	17.	25.5	18.	26.8	18.8	28.1	19.4	29.1	20.2	30.3	21.	31.4	21.8	32.8	22.5	33.6	23.2	34.7	11	20
21	30	17.2	22.1	18.6	25.5	20.4	27.8	22.	30.	24.1	32.	25.5	33.7	26.8	35.6	28.1	37.6	29.1	38.9	30.3	40.4	31.4	42.	32.8	43.4	33.6	44.9	34.7	46.3	21	30
31	40	22.1	28.4	25.5	31.0	27.8	34.1	30.	37.	32.	39.5	33.7	42.	35.6	43.9	37.6	46.	38.9	48.5	40.4	50.5	42.	52.4	43.4	54.2	44.9	56.	46.3	57.8	31	40
41	50	28.4	33.4	31.0	37.4	34.1	41.	37.	43.8	39.5	47.	42.	50.	43.9	52.5	46.	55.6	48.5	58.2	50.5	60.6	52.4	63.	54.2	65.1	56.	67.4	57.8	69.5	41	50
51	60	33.4	38.8	37.4	42.5	41.	48.3	43.8	53.6	47.	55.6	50.	59.	52.5	62.9	55.6	64.7	58.2	67.9	60.6	70.7	63.	73.4	65.1	75.9	67.4	78.5	69.5	81.	51	60
61	70	38.8	44.5	42.5	50.	48.3	55.1	53.6	59.5	55.6	63.2	59.	67.	62.9	71.	64.7	74.4	67.9	77.6	70.7	80.8	73.4	83.8	75.9	86.8	78.5	89.6	81.	92.5	61	70
71	80	44.5	50.8	50.	56.8	55.1	62.3	59.5	66.6	63.2	71.3	67.	75.6	71.	80.	74.4	82.8	77.6	87.5	80.8	91.	83.8	94.5	86.8	97.7	89.6	101.	92.5	104.	71	80
81	90	50.8	56.5	56.8	63.	62.3	68.5	66.6	74.1	71.3	79.2	75.6	82.6	80.	90.3	82.8	93.5	87.5	97.3	91.	101.	94.5	105.	97.7	109.	101.	112.	104.	116.	81	90
91	100	56.5	62.	63.	68.6	68.5	75.3	74.1	81.	79.2	87.7	82.6	93.1	90.3	98.	93.5	103.	97.3	107.	101.	111.	105.	115.	109.	119.	112.	123.	116.	127.	91	100
101	110	62.	66.6	68.6	75.	75.3	81.	81.	89.6	87.7	95.7	93.1	102.	98.	107.	103.	112.	107.	117.	111.	121.	115.	126.	119.	131.	123.	135.	127.	139.	101	110
111	120	66.6	69.4	75.	78.5	81.	89.6	89.6	98.5	95.7	104.	102.	110.	107.	116.	112.	122.	117.	126.	121.	131.	126.	136.	131.	141.	135.	146.	139.	151.	111	120
121	130	69.4	74.5	78.5	87.9	89.6	96.3	98.5	104.	104.	111.	110.	118.	116.	124.	122.	130.	126.	136.	131.	141.	136.	147.	141.	152.	146.	157.	151.	162.	121	130
131	140	74.5	81.	87.9	90.6	96.3	100.	104.	108.	111.	116.	118.	123.	124.	129.	130.	133.	136.	145.	141.	152.	147.	157.	152.	163.	157.	168.	162.	174.	131	140
141	150	81.	90.1	90.6	100.1	100.	110.	108.	119.	116.	127.	123.	133.	129.	140.	133.	147.	145.	155.	152.	162.	157.	168.	163.	174.	168.	179.	174.	185.	141	150
151	160	90.1	96.	100.1	106.5	110.	117.	119.	127.	127.	132.	133.	141.	140.	149.	147.	156.	155.	165.	162.	172.	168.	179.	174.	185.	179.	191.	185.	197.	151	160
161	170	96.	101.5	106.5	112.	117.	124.	127.	131.	132.	140.	141.	149.	149.	158.	156.	166.	165.	175.	172.	182.	179.	189.	185.	196.	191.	202.	197.	209.	161	170
½	....	102.	128.	114.	138.	125.	152.	131.	166.	141.	178.	150.	190.	159.	200.	166.	210.	175.	221.	182.	230.	189.	239.	196.	247.	.....	.....	.....	.....	½	....
⅝	....	128.	151.	138.	172.	152.	189.	166.	204.	178.	220.	190.	234.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅝	....
⅞	....	151.	184.	172.	208.	189.	228.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅞	....
1⅛	....	184.	219.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1⅛	....

		18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	36	40	40	43	43	46	46	50	50	55	55	60		
Two	0-10	8.7	12.3	9.1	12.9	9.6	13.5	10.	14.1	10.3	14.6	10.7	15.3	11.3	15.9	11.8	16.9	12.4	17.6	12.9	18.3	13.4	18.9	14.	19.8	14.7	20.7	15.3	21.6	Two	0-10
0	10	12.3	24.6	12.9	25.8	13.5	27.	14.1	28.2	14.6	29.2	15.3	30.4	15.9	31.8	16.9	33.8	17.6	35.2	18.3	36.6	18.9	37.8	19.8	39.5	20.7	41.5	21.6	43.3	0	10
11	20	24.6	36.8	25.8	38.6	27.	40.5	28.2	42.7	29.2	43.9	30.4	45.5	31.8	47.7	33.8	50.5	35.2	52.6	36.6	55.7	37.8	56.6	39.5	59.1	41.5	62.	43.3	64.9	11	20
21	30	36.8	49.	38.6	51.5	40.5	54.	42.7	56.2	43.9	58.5	45.5	61.6	47.7	63.6	50.5	67.5	52.6	70.2	55.7	73.	56.6	75.5	59.1	78.9	62.	82.6	64.9	86.5	21	30
31	40	49.	61.1	51.5	64.3	54.	67.3	56.2	70.2	58.5	73.	61.6	75.6	63.6	79.5	67.5	84.2	70.2	87.6	73.	91.2	75.5	94.3	78.9	98.5	82.6	103.	86.5	108.	31	40
41	50	61.1	73.5	64.3	77.2	67.3	81.	70.2	84.4	73.	87.6	75.6	91.	79.5	95.5	84.2	101.	87.6	105.	91.2	110.	94.3	113.	98.5	118.	103.	124.	108.	130.	41	50
51	60	73.5	85.5	77.2	90.	81.	94.2	84.4	99.3	87.6	102.	91.	106.	95.5	111.	101.	118.	105.	123.	110.	127.	113.	132.	118.	138.	124.	145.	130.	151.	51	60
61	70	85.5	98.	90.	103.	94.2	108.	99.3	112.	102.	117.	106.	121.	111.	127.	118.	135	123.	140.	127.	146.	132.	151.	138.	158.	145.	165.	151.	173.	61	70
71	80	98.	110.	103.	116.	108.	122.	112.	127.	117.	132.	121.	137.	127.	143.	135.	152.	140.	158.	146.	165.	151.	170.	158.	178.	165.	186.	173.	195.	71	80
81	90	110.	123.	116.	129.	122.	135.	127.	141.	132.	146.	137.	152.	143.	159.	152.	169.	158.	176.	165.	183.	170.	189.	178.	198.	186.	207.	195.	216.	81	90
91	100	123.	135.	129.	142.	135.	148.	141.	154.	146.	160.	152.	167.	159.	175.	169.	185.	176.	193.	183.	205.	189.	208.	198.	217.	.....	.....	.....	.....	91	100
101	110	135.	147.	142.	154.	148.	162.	154.	169.	160.	175.	167.	182.	175.	191.	185.	202.	193.	211.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	101	110
111	120	147.	159.	154.	167.	162.	176.	169.	183.	175.	190.	182.	197.	191.	207.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	111	120
121	130	159.	171.	167.	180.	176.	188.	183.	197.	190.	204.	197.	212.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	121	130
131	140	171.	184.	180.	193.	188.	202.	197.	211.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	131	140
141	150	184.	196.	193.	206.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	141	150
151	160	196.	210.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	151	160
161	170	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	161	170
½	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	½	....
⅝	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅝	....
⅞	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅞	....
1⅛	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1⅛	....

TABLE 6—CAPACITIES IN SQUARE FEET OF DIRECT RADIATION OF ORIFICES DESIGNED FOR VAPOR SYSTEMS—*Continued*

Vapor Rating of Orifice	Section 3										Height Above Midpoint of Heater in Feet										Section 3		Vapor Rating of Orifice
	4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18	
Two 0–10																													Two 0–10
0 10	4.2	8.5	4.8	9.5	5.2	10.5	5.7	11.4	6.1	12.2	6.5	13.0	6.8	13.7	7.2	14.4	7.3	14.6	7.6	15.2	8.	15.8	8.2	16.5	8.5	17.1	8.8	17.6	0 10
11 20	8.5	12.7	9.5	14.3	10.5	15.3	11.4	16.7	12.2	18.0	13.0	19.2	13.7	20.2	14.4	21.0	14.6	21.9	15.2	22.8	15.8	23.8	16.5	24.7	17.1	25.5	17.6	26.3	11 20
21 30	12.7	16.3	14.3	18.4	15.3	20.4	16.7	21.5	18.0	24.0	19.2	25.6	20.2	27.0	21.0	28.4	21.9	29.1	22.8	30.4	23.8	31.6	24.7	32.9	25.5	34.	26.3	35.1	21 30
31 40	16.3	20.2	18.4	22.0	20.4	25.6	21.5	27.8	24.0	29.9	25.6	31.7	27.0	33.2	28.4	34.9	29.1	36.4	30.4	38.0	31.6	39.6	32.9	41.1	34.	42.5	35.1	43.8	31 40
41 50	20.2	24.4	22.0	27.7	25.6	30.6	27.8	33.2	29.9	35.1	31.7	37.5	33.2	39.8	34.9	42.0	36.4	43.6	38.0	45.6	39.6	47.5	41.1	49.4	42.5	51.	43.8	52.6	41 50
51 60	24.4	28.4	27.7	32.1	30.6	34.8	33.2	38.0	35.1	41.9	37.5	43.6	39.8	44.6	42.0	49.2	43.6	50.9	45.6	53.1	47.5	55.4	49.4	57.5	51.	59.5	52.6	61.4	51 60
61 70	28.4	32.5	32.1	35.7	34.8	39.7	38.0	43.5	41.9	47.4	43.6	50.5	44.6	53.5	49.2	56.4	50.9	58.2	53.1	60.9	55.5	63.4	57.5	65.6	59.5	68.	61.4	70.2	61 70
71 80	32.5	34.9	35.7	40.1	39.7	43.6	43.5	49.5	47.4	53.4	50.5	56.9	53.5	60.2	56.4	63.2	58.2	65.6	60.9	69.5	63.4	71.4	65.6	74.0	68.	76.6	70.2	78.	71 80
81 90	34.9	38.8	40.1	43.6	43.6	50.4	49.5	55.0	53.4	59.2	56.9	63.2	60.2	66.4	63.2	69.8	65.6	73.0	69.5	76.1	71.4	79.4	74.0	82.4	76.6	85.1	78.	88.	81 90
91 100	38.8	42.6	43.6	49.6	50.4	55.4	55.0	60.3	59.2	64.1	63.2	68.5	66.4	72.6	69.8	76.5	73.0	80.0	76.1	83.6	79.4	87.0	82.4	90.4	85.1	93.5	88.	96.	91 100
101 110	42.6	47.6	49.6	54.4	55.4	60.5	60.3	65.1	64.1	70.2	68.5	75.0	72.6	79.5	76.5	83.5	80.0	87.4	83.6	91.2	87.0	95.0	90.4	98.5	93.5	102.	96.5	105.	101 110
111 120	47.6	51.5	54.4	58.7	60.5	64.1	65.1	70.2	70.2	75.8	75.0	81.0	79.5	83.5	83.5	91.0	87.4	94.6	91.2	99.0	95.0	106.	98.5	107.	102.	111.	105.	114.	111 120
121 130	51.5	55.5	58.7	63.2	64.1	68.1	70.2	75.8	75.8	81.0	81.0	88.2	83.5	93.4	91.0	98.0	94.6	104.	99.0	106.	106.	111.	107.	115.	111.	119.	114.	123.	121 130
131 140	55.5	59.5	63.2	66.5	68.1	74.1	75.8	81.0	81.0	88.6	88.2	94.5	93.4	100.	98.0	105.	104.	109.	106.	114.	111.	119.	115.	123.	119.	127.	123.	131.	131 140
141 150	59.5	63.2	66.5	71.0	74.1	79.2	81.0	87.9	88.6	94.5	94.5	101.	100.	107.	105.	112.	109.	116.	114.	122.	119.	127.	123.	132.	127.	136.	131.	140.	141 150
151 160	63.2	65.5	71.0	75.5	79.2	82.5	87.9	93.4	94.5	100.	101.	107.	107.	113.	112.	116.	116.	124.	122.	129.	127.	135.	132.	140.	136.	145.	140.	149.	151 160
161 170	65.5	69.5	75.5	80.0	82.5	84.8	93.4	99.0	100.	106.	107.	113.	113.	120.	116.	126.	124.	131.	129.	137.	135.	143.	140.	148.	145.	153.	149.	158.	161 170
½	69.5	90.1	80.0	103.	84.8	115.	99.0	125.	106.	131.	113.	139.	120.	147.	126.	151.	131.	166.	137.	173.	143.	180.	148.	187.	153.	194.	158.	200.	½ ....
⅝	90.1	111.	103.	127.	115.	134.	125.	147.	131.	157.	139.	172.	147.	183.	151.	194.	166.	205.	173.	214.	180.	222.	187.	231.	194.	239.	.....	.....	⅝ ....
⅝	111.	134.	127.	143.	134.	162.	147.	178.	157.	193.	172.	207.	183.	220.	194.	232.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅝ ....
1¼	134.	145.	143.	172.	162.	193.	178.	213.	193.	230.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1¼ ....
¾	145.	200.	172.	234.	193.	263.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	¾ ....

	18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	36	40	40	43	43	46	46	50	50	55	55	60	
Two 0–10			7.	9.8	7.3	10.3	7.6	10.8	7.9	11.2	8.2	11.6	8.7	12.2	9.1	12.8	9.6	13.5	10.	14.1	10.3	14.6	10.8	15.2	11.3	16.	13.5	16.8	Two 0–10
0 10	9.3	18.6	9.8	19.6	10.3	20.8	10.8	21.5	11.2	22.4	11.6	23.2	12.2	24.3	12.8	25.6	13.5	27.1	14.1	28.2	14.6	29.2	15.2	30.4	16.	32.	16.8	33.5	0 10
11 20	18.6	27.9	19.6	29.4	20.8	30.9	21.5	32.3	22.4	33.5	23.2	34.8	24.3	36.6	25.6	38.4	27.1	40.7	28.2	42.1	29.2	43.7	30.4	45.7	32.	48.	33.5	50.1	11 20
21 30	27.9	37.2	29.4	39.2	30.9	41.2	32.3	43.8	33.5	44.6	34.8	46.5	36.6	48.8	38.4	51.2	40.7	54.1	42.1	56.2	43.7	58.2	45.7	60.8	48.	63.8	50.1	66.9	21 30
31 40	37.2	46.4	39.2	49.	41.2	51.5	43.8	53.8	44.6	55.8	46.5	58.0	48.8	61.	51.2	63.8	54.1	67.6	56.2	70.2	58.2	72.6	60.8	76.	63.8	79.7	66.9	83.5	31 40
41 50	46.4	55.7	49.	59.	51.5	61.8	53.8	64.6	55.8	67.0	58.0	69.6	61.	73.3	63.8	76.7	67.6	81.2	70.2	84.4	72.6	86.4	76.	91.3	79.7	95.	83.5	100.	41 50
51 60	55.7	65.	59.	68.5	61.8	72.	64.6	75.2	67.	78.	69.6	81.1	73.3	85.4	76.7	89.4	81.2	94.5	84.4	98.1	86.4	102.	91.3	106.	95.	112.	100.	117.	51 60
61 70	65.	74.4	68.5	78.4	72.	82.4	75.2	86.0	78.	89.4	81.1	92.8	85.4	97.6	89.4	102.	94.5	108.	98.1	112.	102.	116.	106.	122.	112.	128.	117.	134.	61 70
71 80	74.4	83.6	78.4	83.3	82.4	92.8	86.	97.	89.4	100.	92.8	105.	97.6	110.	102.	115.	108.	122.	112.	127.	116.	131.	122.	137.	128.	143.	134.	151.	71 80
81 90	83.6	93.	88.3	98.1	92.8	103.	97.	108.	100.	112.	105.	116.	110.	122.	115.	128.	122.	136.	127.	141.	131.	146.	137.	152.	143.	160.	151.	168.	81 90
91 100	93.	102.	98.1	108.	103.	113.	108.	118.	112.	123.	116.	127.	122.	134.	128.	141.	136.	150.	141.	154.	146.	160.	152.	167.	160.	176.	168.	184.	91 100
101 110	102.	112.	108.	117.	113.	124.	118.	129.	123.	134.	127.	139.	134.	147.	141.	154.	150.	162.	154.	169.	160.	175.	167.	182.	176.	191.	184.	200.	101 110
111 120	112.	121.	117	127.	124.	134.	129.	140.	134.	145.	139.	151.	147.	159.	154.	166.	162.	176.	169.	183.	175.	189.	182.	198.	191.	208.	.....	.....	111 120
121 130	121.	130.	127.	137.	134.	144.	140.	150.	145.	156.	151.	162.	159.	171.	166.	179.	176.	189.	183.	196.	189.	204.	198.	212.	.....	.....	.....	.....	121 130
131 140	130.	139.	137.	147.	144.	155.	150.	161.	156.	168.	162.	174.	171.	183.	179.	192.	189.	203.	196.	211.	.....	.....	.....	.....	.....	.....	.....	.....	131 140
141 150	139.	149.	147.	157.	155.	165.	161.	172.	168.	179.	174.	185.	183.	195.	192	204.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	141 150
151 160	149.	158.	157.	167.	165.	175.	172.	183.	179.	190.	185.	199.	195.	208.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	151 160
161 170	158.	167.	167.	177.	175.	185.	183.	194.	190.	201.	199.	209.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	161 170
½	167.	212.	177.	223.	185.	234.	194.	245.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	½ ....
⅝	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅝ ....
⅝	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅝ ....
1¼	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1¼ ....
¾	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	¾ ....

Vapor Rating of Orifice	Section 4						Height Above Midpoint of Heater in Feet						Secton 4		Vapor Rating of Orifice
	4½–5	5–6	6–7	7–8	8–9	9–10	10–11	11–12	12–13	13–14	14–15	15–16	16–17	17–18	
0–10	2.9– 5.8	3.3– 6.7	3.7– 7.4	4.0– 8.1	4.3– 8.7	4.6– 9.3	4.9– 9.8	5.1– 10.4	5.2– 10.5	5.3– 10.7	5.5– 10.9	5.7– 11.4	5.9– 11.9	6.1– 12.3	0–10
11–20	5.8– 8.7	6.7– 10.0	7.4– 11.1	8.1– 12.1	8.7– 13.0	9.3– 13.9	9.8– 14.7	10.4– 15.0	10.5– 15.2	10.7– 15.6	10.9– 16.3	11.4– 17.0	11.9– 17.7	12.3– 18.4	11–20
21–30	8.7– 10.7	10.0– 13.3	11.1– 14.8	12.1– 15.3	13.0– 16.5	13.9– 17.7	14.7– 18.9	15.0– 20.0	15.2– 20.2	15.6– 20.8	16.3– 21.8	17.0– 22.8	17.7– 23.6	18.4– 24.5	21–30
31–40	10.7– 14.5	13.3– 15.4	14.8– 17.2	15.3– 19.0	16.5– 20.6	17.7– 21.0	18.9– 22.6	20.0– 25.6	20.2– 24.7	20.8– 25.9	21.8– 27.2	22.8– 28.4	23.6– 29.5	24.5– 30.6	31–40
41–50	14.5– 15.5	15.4– 18.1	17.2– 20.7	19.0– 21.3	20.6– 23.4	21.0– 26.8	22.6– 28.3	25.6– 30.0	24.7– 30.3	25.9– 31.2	27.2– 32.7	28.4– 34.0	29.5– 35.4	30.6– 36.8	41–50
51–60	15.5– 18.1	18.1– 21.0	20.7– 22.1	21.3– 26.4	23.4– 28.8	26.8– 31.0	28.3– 33.0	30.0– 33.8	30.3– 34.7	31.2– 36.4	32.7– 38.1	34.0– 40.2	35.4– 41.3	36.8– 42.9	51–60
61–70	18.1– 20.7	21.0– 24.2	22.1– 27.6	26.4– 32.0	28.8– 33.0	31.0– 34.1	33.0– 36.4	33.8– 38.6	34.7– 39.6	36.4– 41.7	38.1– 43.5	40.2– 45.4	41.3– 47.3	42.9– 49.0	61–70
71–80	20.7– 23.4	24.2– 27.4	27.6– 31.0	32.0– 33.2	33.0– 35.6	34.1– 38.4	36.4– 41.0	38.6– 43.6	39.6– 44.7	41.7– 47.0	43.5– 49.0	45.4– 51.2	47.3– 53.3	49.0– 55.2	71–80
81–90	23.4– 26.0	27.4– 30.6	31.0– 33.2	33.2– 36.2	35.6– 39.6	38.4– 42.7	41.0– 44.4	43.6– 47.0	44.7– 49.7	47.0– 52.3	49.0– 54.5	51.2– 57.0	53.3– 59.2	55.2– 61.4	81–90
91–100	26.0– 28.4	30.6– 33.2	33.2– 35.5	36.2– 39.6	39.6– 43.5	42.7– 45.2	44.4– 51.0	47.0– 54.0	49.7– 54.5	52.3– 57.4	54.5– 59.7	57.0– 62.4	59.2– 65.0	61.4– 67.4	91–100
101–110	28.4– 31.0	33.2– 33.6	35.5– 38.7	39.6– 43.4	43.5– 45.8	45.2– 52.1	51.0– 55.6	54.0– 58.9	54.5– 59.4	57.4– 62.6	59.7– 65.2	62.4– 68.0	65.0– 71.0	67.4– 73.5	101–110
111–120	31.0– 33.0	33.6– 36.4	38.7– 41.9	43.4– 44.5	45.8– 52.6	52.1– 56.6	55.6– 60.4	58.9– 63.2	59.4– 64.5	62.6– 68.0	65.2– 71.0	68.0– 73.8	71.0– 77.0	73.5– 80.0	111–120
121–130	33.0– 34.0	36.4– 39.1	41.9– 43.6	44.5– 51.9	52.6– 56.5	56.6– 60.8	60.4– 63.5	63.2– 67.4	64.5– 69.3	68.0– 73.0	71.0– 76.0	73.8– 79.4	77.0– 82.6	80.0– 85.6	121–130
131–140	34.0– 35.0	39.1– 41.8	43.6– 45.4	51.9– 55.4	56.5– 60.5	60.8– 63.6	63.5– 68.1	67.4– 72.6	69.3– 74.3	73.0– 78.8	76.0– 81.4	79.4– 85.0	82.6– 88.7	85.6– 91.9	131–140
141–150	35.0– 36.6	41.8– 43.6	45.4 53.4	55.4– 59.3	60.5– 63.2	63.6– 68.0	68.1– 72.6	72.6– 77.1	74.3– 79.2	78.0– 83.5	81.4– 87.0	85.0– 91.0	88.7– 94.5	91.9– 98.0	141–150
151–160	36.6– 38.8	43.6– 45.0	53.4– 55.0	59.3– 58.4	63.2– 64.4	68.0– 70.0	72.6– 75.2	77.1– 80.0	79.2– 84.3	83.5– 88.0	87.0– 92.0	91.0– 96.0	94.5–101.	98.0–106.	151–160
161–170	38.8– 41.2	45.0– 46.1	55.0– 60.1	58.4– 64 5	64.4– 70.8	70.0– 76.4	75.2– 81.9	80.0– 82.3	84.3– 89.4	88.0– 94.0	92.0– 98.0	96.0– 102.	101.–106.	106.–110.	161–170
½"	41.2– 45.2	46.1– 64.0	60.1– 72.8	64.5– 81.5	70.–8 91.8	76.4– 98.5	81.9–105.	82.3–111.	89.4–113.	94.0–118.	98.0–124.	102. –131.	106.– 137.	110.–143.	½"
⅝"	45.2– 63.2	64.0– 77.6	72.8– 79.0	81.5–104.	91.8–113.	98.5–123.	105. –130.	111. –136.	113. –138.	118. –147.	124. –153.	131. –160.	137. –166.	143. –172.	⅝"
⅝"	63.2– 76.2	77.6– 95.2	79.0–112.	104. –125.	113. –136.	123. –137.	130. –146.	136. –157.	138. –168.	147. –177.	153. –184.	160. –193.	166. –201.	172. –208.	⅝"
1¼"	76.2– 97.2	95.2–117.	112. –134.	125. –133.	136. –148.	137. –163.	146. –176.	157. –187.	168. –201.	177. –211.	184. –221.	193. –231.	.....	.....	1¼"
¾"	97.2–132.	117. –131.	134. –158.	133. –184.	148. –201.	163. –220.	176. –238.	187. –254.	.....	.....	.....	.....	.....	.....	¾"
⅞"	132.– 152.	131. –191.	158. –210.	184. –240.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅞"
1"	152.– 191.	191. –252.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1"

	18–20	20–22	22–24	24–26	26–28	28–30	30–33	33–36	36–40	40–43	43–46	46–50	50–55	55–60	
0–10	6.5– 13.1	6.9– 13.8	7.3– 14.6	7.7– 15.3	8.0– 16.0	8.3– 16.5	8.8– 17.5	9.2– 18.4	9.8– 19.6	10.1– 20.7	10.5– 21.0	11.1– 22.0	11.6– 23.1	12.1– 24.1	0–10
11–20	13.1– 19.6	13.8– 20.7	14.6– 21.8	15.3– 23.0	16.0– 23.9	16.5– 24.8	17.5– 26.1	18.4– 27.4	19.6– 29.2	20.7– 30.2	21.0– 31.4	22.0– 33.0	23.1– 34.6	24.1– 36.1	11–20
21–30	19.6–26.5	20.7– 27.6	21.8– 29.1	23.0– 30.6	23.9– 31.9	24.8– 33.0	26.1– 35.0	27.4– 36.6	29.2– 39.0	30.2– 40.5	31.4– 42.0	33.0– 44.0	34.6– 46.1	36.1– 48.2	21–30
31–40	26.5–32.5	27.6– 35.5	29.1– 36.4	30.6– 38.2	31.9– 39.8	33.0– 41.3	35.0– 43.5	36.6– 45.7	39.0– 48.7	40.5– 50.5	42.0– 52.3	44.0– 54.8	46.1– 57.6	48.2– 60.2	31–40
41–50	32.5– 39.2	35.5– 41.5	36.4– 43.8	38.2– 45.8	39.8– 48.0	41.3– 49.5	43.5– 52.3	45.7– 55.0	48.7– 58.5	50.5– 60.5	52.3– 63.0	54.8– 66.0	57.6– 69.2	60.2– 72.5	41–50
51–60	39.2– 45.6	41.5– 48.3	43.8– 51.0	45.8– 53.4	48.0– 55.7	49.5– 57.7	52.3– 61.0	55.0– 64.0	58.5– 68.1	60.5– 70.6	63.0– 73.3	66.0– 76.9	69.2– 80.5	72.5– 84.5	51–60
61–70	45.6– 52.1	48.3– 55.4	51.0– 58.2	53.4– 61.0	55.7– 63.8	57.7– 66.0	61.0– 68.6	64.0– 7.22	68.1– 76.0	70.6– 80.7	73.3– 84.0	76.9– 87.8	80.5– 92.3	84.5– 97.2	61–70
71–80	52.1– 58.7	55.4– 62.3	58.2– 65.5	61.0– 68.9	63.8– 72.0	66.0– 74.5	68.6– 76.1	72.2– 82.5	76.0– 87.8	80.7– 91.0	84.0– 94.5	87.8– 99.0	92,3–104.	97.2–109.	71–80
81–90	58.7– 65.4	62.3– 69.3	65.5– 73.0	68.9– 76.5	72.0– 80.0	74.5– 82.9	76.1– 85.0	82.5– 91.8	87.8– 97.5	91.0–101.0	94.5–105.	99.0–110.	104. –116.	109. –121.	81–90
91–100	65.4– 71.6	69.3– 76.0	73.0– 80.0	76.5– 84.5	80.0– 87.6	82.9– 91.0	85.0– 96.0	91.8–101.	97.5–107.	101.0–111.	105. –113.	110. –121.	116. –127.	121. –133.	91–100
101–110	71.6– 78.2	76.0– 83.0	80.0– 87.3	84.5– 91.6	87.6– 95.6	91.0– 99.1	96.0–104.	101. –110.	107. –117.	111. –121.	116. –126.	121. –128.	127. –136.	133. –145.	101–110
111–120	78.2– 85.0	83.0– 90.0	87.3– 94.8	91.6– 99.5	95.6–104.	99.1–109.	104. –114.	110. –119.	117. –126.	121. –131.	126. –136.	128. –143.	136. –150.	145. –157.	111–120
121–130	85.0– 91.0	90.0– 96.8	94.8–102.	99.5–107.	104. –112.	109. –116.	114. –121.	119. –128.	126. –135.	131. –141.	136. –147.	143. –154.	150. –161.	157. –169.	121–130
131–140	91.0– 97.9	96,8–104.	102. –109.	107. –114.	112. –119.	116. –124.	121. –130.	128. –137.	135. –146.	141. –151.	147. –158.	154. –165.	161. –173.	169. –181.	131–140
141–150	97.9–104.	104. –110.	109. –116.	114. –122.	119. –128.	124. –132.	130. –140.	137. –147.	146. –155.	151. –161.	158. –168.	165. –175.	173. –184.	181. –193.	141–150
151–160	104. –111.	110. –118.	116. –124.	122. –130.	128. –135.	132. –140.	140. –149.	147. –155.	155. –164.	161. –171.	168. –179.	175. –187.	184. –196.	193. –205.	151–160
161–170	111. –117.	118. –124.	124. –131.	130. –137.	135. –144.	140. –148.	149. –156.	155. –165.	164. –173.	171. –181.	179. –189.	187. –198.	196. –207	.....	161–170
½"	117. –150.	124. –158.	131. –166	137. –174.	144. –182.	148.– 190.	156. –200.	165. –210.	173. -.220	131. -229.	189. –239.	198. –250.	.....	.....	½"
⅝"	150. –183.	158. –194.	166. –202	174. –214.	182. –224.	190. –231.	.....	.....	.....	.....	.....	.....	.....	.....	⅝"
⅝"	183. –220.	194 –235.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅝"
1¼"	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1¼"
¾"	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	¾"
⅞"	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	⅞"
1"	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1"

just considered. It differs from it in that it has second-floor radiators and a branched circuit. In the case of a branched circuit of this sort it is probably best to arrange the mains one above the other with the flow mains on top, as illustrated in Fig. 7, instead of side by side as is the usual practice. If this is not practical, the shorter branch may be carried over the longer one by means of two 45 deg. bends as illustrated in Fig. 8. If head room is important, as it usually is, the *Y* connections may be rotated downward until the upper pipes lie flat upon the lower pipes.

It will also be noticed that in some cases the construction of the building is such that the lateral portions of the risers require an extra turn. In cases like this, illustrated by radiators 2 and 6, it is best to go up with a 45 deg. elbow instead of a 90 deg. elbow, as illustrated by Fig. 9, or else make the turn with a 45 deg. elbow as illustrated in Fig. 10. In either case, use one 90 deg. elbow and one 45 deg. elbow, where two 90 deg. elbows would otherwise be required. An even better arrangement, when it can be used, is to eliminate the bend and run the lateral portions of the risers so that one is directly above the other as illustrated in Fig. 4B. For a system of this size it is probably best to arrange the data as shown in Table 8.

The numbers in column 1 of Table 8 give the radiator numbers consecutively beginning at the end of the branch farthest from the heater. The numbers in column 2 give the section of the line in which the radiator is located. The numbers in column 3 represent the height of the mid-point of each radiator above the mid-point of the heater. Thus, Radiator 11 is 8 ft. 0 in. above the mid-point of the heater, while Radiator 10 is 7 ft. 4 in. above the mid-point of the heater. The heights vary a trifle even among radiators on the same floor because some of the radiators are taller than others. The numbers in

TABLE 7—SETTING OF VALVES CALIBRATED IN SQUARE FEET OF RADIATION

Square Feet of Radiation	Section 1 — Height Above Midpoint of Heater in Feet																												Section 1	Square Feet of Radiation
	4½–5	5–6	6–7	7–8	8–9	9–10	10–11	11–12	12–13	13–14	14–15	15–16	16–17	17–18	18–20	20–22	22–24	24–26	26–28	28–30	30–33	33–36	36–40	40–43	43–46	46–50	50–55	55–60		
10	13	12	11	10	10	9	9																							10
15	20	18	16	15	14	14	13	13	12	12	11	11	11	10	10	9	9	8												15
20	27	24	22	21	20	19	18	17	16	16	15	15	14	14	13	12	12	11	11	11	10	10	9	9	9					20
25	33	30	28	26	24	23	22	21	20	19	18	18	18	17	16	15	15	14	14	13	13	12	11	11	11	10	10	9		25
30	40	36	33	31	29	28	26	25	24	23	22	22	21	21	19	18	18	17	17	16	15	15	14	13	13	12	12	11		30
35	47	43	39	36	34	32	30	29	28	27	26	25	25	24	23	22	21	20	19	19	18	17	16	15	15	14	14	13		35
40	54	49	45	42	39	37	35	34	32	31	29	29	28	27	26	25	24	23	22	21	20	19	18	17	17	16	16	15		40
45	61	55	51	47	44	42	40	38	36	35	33	33	32	31	29	28	26	25	25	24	23	22	21	19	19	18	18	17		45
50	67	60	56	52	49	46	44	42	41	39	37	36	35	34	32	31	30	28	28	27	25	24	23	22	21	20	19	19		50
60	80	73	67	62	58	55	53	50	49	47	44	44	42	41	39	37	35	34	33	32	30	29	27	26	26	25	23	22		60
70	93	85	78	73	68	65	62	59	57	55	52	51	49	48	45	43	41	40	39	37	35	34	32	30	30	29	27	26		70
80	107	97	85	83	78	74	70	67	65	62	59	58	56	55	52	49	47	45	44	43	41	39	37	35	34	33	31	30		80
90	120	109	100	93	88	83	79	76	73	70	66	65	63	62	58	55	53	51	50	48	46	44	41	39	38	37	35	34		90
100	133	121	111	104	97	92	88	84	81	78	74	73	70	68	65	63	59	57	55	53	51	48	46	43	43	41	39	37		100
110	147	133	122	114	107	101	97	92	89	86	81	80	77	75	71	68	65	62	61	58	56	53	50	47	47	45	43	41		110
120	160	145	134	125	117	111	106	101	97	94	88	87	84	82	78	74	71	68	66	64	61	58	55	52	51	49	47	45		120
130	174	157	145	135	127	120	114	109	105	101	96	94	91	89	84	80	77	74	72	69	66	63	60	56	55	53	51	48		130
140	191	173	159	148	138	131	124	119	113	109	103	102	98	96	91	86	83	79	77	74	71	68	64	60	60	57	55	52		140
150	205	185	170	158	148	140	133	126	123	117	110	109	105	103	97	92	89	85	83	80	76	73	69	65	64	61	58	56		150
160	218	197	181	168	158	149	142	135	130	125	118	116	113	109	104	99	94	91	88	85	81	77	73	69	68	65	62	60		160
180	245	221	203	189	177	168	159	152	146	140	133	131	127	123	116	111	107	102	99	96	91	87	82	78	77	74	70	67		180
200	273	246	226	210	197	187	177	169	162	156	147	145	141	137	129	123	118	113	110	106	101	97	92	86	85	82	78	75		200

Square Feet of Radiation	Section 2 — Height Above Midpoint of Heater in Feet																												Section 2	Square Feet of Radiation
	4½–5	5–6	6–7	7–8	8–9	9–10	10–11	11–12	12–13	13–14	14–15	15–16	16–17	17–18	18–20	20–22	22–24	24–26	26–28	28–30	30–33	33–36	36–40	40–43	43–46	46–50	50–55	55–60		
10	17	15	14	13	12	12	11	11	10	10	9	9	9																	10
15	26	23	21	20	18	17	17	16	15	15	14	14	13	13	12	12	11	11	10	10	9	9	9							15
20	35	31	29	27	25	23	22	21	20	20	19	18	18	17	16	15	15	14	14	13	13	12	11	11	11	10	10	9		20
25	44	39	36	33	31	29	28	27	25	25	24	23	22	21	20	19	19	18	17	16	16	15	14	14	13	13	12	12		25
30	52	47	43	40	37	35	33	32	30	30	28	28	27	26	24	23	22	21	20	20	19	18	17	16	16	15	14	14		30
35	63	56	51	47	44	41	39	38	35	35	33	32	31	30	28	27	26	25	24	23	22	21	20	19	18	18	17	16		35
40	71	64	58	54	51	47	45	43	41	39	38	37	35	34	33	31	29	28	27	26	25	24	23	22	21	20	19	18		40
45	83	73	66	61	57	54	51	49	46	44	43	41	40	39	36	35	33	32	31	30	28	27	26	25	24	23	22	21		45
50	88	79	72	68	62	59	56	53	51	50	48	46	44	43	41	39	38	35	34	33	31	30	28	27	26	25	24	23		50
60	106	95	87	80	75	71	67	64	61	59	57	55	53	52	49	47	44	42	41	39	38	36	34	33	32	30	29	28		60
70	126	112	102	94	88	83	78	75	72	69	66	64	62	60	57	54	52	50	48	46	44	42	40	38	37	35	34	32		70
80	144	128	116	107	100	95	90	86	82	79	76	74	71	69	65	62	59	57	54	53	50	48	45	44	42	40	38	37		80
90	159	142	130	120	112	106	101	96	92	89	85	83	80	77	73	70	66	64	61	59	56	54	51	49	47	45	43	41		90
100	177	158	144	133	125	118	112	106	100	99	95	92	89	86	81	78	74	71	68	66	63	60	57	55	53	50	48	46		100
110	195	174	159	147	138	130	123	117	113	109	105	101	98	95	90	85	81	78	75	73	69	66	62	60	58	56	53	51		110
120	212	190	173	160	150	142	134	128	123	119	114	110	106	103	98	93	89	85	82	79	75	72	68	66	63	61	58	55		120
130	230	206	188	173	163	153	145	138	133	128	123	120	115	112	106	101	96	92	89	85	82	78	74	71	69	66	63	60		130
140	260	230	208	192	179	168	160	152	143	138	133	128	124	120	114	110	104	99	95	92	88	83	79	76	74	71	67	64		140
150	279	246	223	206	192	180	172	163	154	148	142	138	133	129	122	116	111	106	102	99	94	89	85	82	78	76	72	69		150
160	296	262	238	219	214	192	182	173	164	158	152	147	142	138	130	124	118	114	109	105	100	96	91	87	85	81	77	74		160
180	332	294	267	246	230	216	204	195	184	178	171	165	160	155	146	140	133	128	123	119	113	107	102	98	95	91	87	83		180
200									225	197	190	184	178	172	163	155	148	142	136	132	126	119	113	109	106	101	96	92		200

column 4 of the tabulation represent the number of square feet of radiation furnished by each radiator. Number 11, for instance, has 160 sq. ft., No. 10 has 24 sq. ft. and so on down the column.

The numbers in column 5 represent the number of square feet of radiation that the section of the mains immediately preceding the radiator must carry. Thus, the mains immediately preceding No. 11 must carry 160 sq. ft., the section preceding No. 10, 184 sq. ft. and that preceding No. 9, 249 sq. ft. Each number in this column is obtained by adding together the number on its left and the number immediately above it. Thus, 184 is obtained by adding 24 to 160, 249 is obtained by adding 65 to 184, and 270 is obtained by adding 21 to 249. The only exceptions to this are 676 and 981. These represent the number of square feet that must be carried by a section immediately preceding a junction. They are obtained by adding together the numbers of square feet carried by each branch. Thus, 676 is obtained by adding 208 to 468, and 981 is obtained by adding 230 to 751.

The numbers in columns 6 and 7 are riser sizes. They are obtained by selecting the sizes from Table 1 that correspond to the number of square feet for each radiator as given in column 4. The numbers in columns 8 and 9 are the sizes of the mains for the sections whose capacities are given in column 5. In this system the height from the mid-point of the heater to the mid-point of the lowest portion of the mains is 3 ft. 6 in. Therefore, Table 2 is used to determine the sizes of the mains.

The length of a branched line must always be measured along the longest branch; consequently the length of the line in this case is 3 ft. + 15 ft. + 31½ ft. + 4½ ft. + 8¼ ft. + 10 ft. or 72¼ ft. Consequently, the proper number of square feet is indicated in column 3 under the column headed "70-80." Thus, for 160 sq.

TABLE 7—SETTING OF VALVES CALIBRATED IN SQUARE FEET OF RADIATION—*Continued*

Square Feet of Radiation	Section 3																												Square Feet of Radiation
	Height Above Midpoint of Heater in Feet																												
	4½–5	5–6	6–7	7–8	8–9	9–10	10–11	11–12	12–13	13–14	14–15	15–16	16–17	17–18	18–20	20–22	22–24	24–26	26–28	28–30	30–33	33–36	36–40	40–43	43–46	46–50	50–55	55–60	
10	24	21	19	17	16	15	15	14	14	13	13	12	12	11	11	10	10												10
15	35	31	28	24	23	23	22	22	20	20	19	18	18	17	16	15	14	14	13	13	12	12	11	11	10	10	9	9	15
20	49	43	39	36	33	31	29	28	27	26	25	24	24	23	21	20	19	18	18	17	16	16	15	14	14	13	12	12	20
25	61	54	49	45	42	39	37	35	34	33	31	30	29	28	27	25	24	23	22	21	20	19	18	18	17	16	16	15	25
30	73	65	58	54	50	47	44	42	41	39	38	36	35	34	32	30	29	28	27	26	25	23	22	21	20	20	19	18	30
35	90	78	70	64	60	56	53	50	48	46	44	42	41	40	37	35	33	32	31	30	29	27	26	25	24	23	22	21	35
40	106	89	80	73	68	64	60	57	55	52	50	48	47	45	43	41	39	37	36	34	33	31	29	28	27	26	25	24	40
45	121	104	92	84	78	73	69	65	61	59	57	54	53	51	48	46	43	42	40	39	37	35	33	32	31	29	28	27	45
50	125	110	99	91	84	79	74	71	68	65	63	61	59	56	53	51	48	48	45	43	41	39	37	35	34	33	31	30	50
60	150	131	119	109	101	95	89	85	82	78	75	73	70	68	64	61	58	56	53	51	49	47	44	42	41	39	37	36	60
70	181	157	141	128	119	112	105	100	96	91	88	85	82	79	75	71	68	65	62	60	57	55	51	50	48	46	44	42	70
80	207	180	161	147	136	127	120	114	110	105	100	97	94	90	85	81	77	74	71	69	65	62	59	57	55	52	50	48	80
90	227	298	178	164	152	144	134	127	123	118	113	109	105	102	96	91	87	83	80	77	73	70	66	64	61	59	56	54	90
100	252	210	198	182	168	158	149	140	137	131	126	122	117	113	107	101	97	93	89	86	82	78	75	71	68	65	62	60	100
110	277	242	218	199	185	174	164	154	151	144	138	133	129	125	117	112	106	102	98	94	90	86	81	78	75	72	69	65	110
120	310	264	238	218	202	190	179	169	164	157	152	145	141	136	128	122	116	111	107	106	98	94	88	85	82	78	75	71	120
130		286	257	236	219	206	194	183	178	170	164	157	152	147	139	132	126	120	116	112	106	101	96	92	89	85	81	77	130
140			294	267	245	229	216	204	191	183	176	170	164	158	150	142	135	130	125	120	114	109	103	99	96	92	87	83	140
150			315	286	264	245	228	219	205	196	189	182	176	170	160	152	145	139	134	128	122	117	110	107	102	98	94	90	150
160				302	279	260	249	232	219	209	202	194	187	181	171	162	155	148	142	137	131	125	117	114	110	105	100	95	160
180					315	298	271	262	256	235	226	218	211	204	192	173	174	167	160	154	147	140	132	128	123	118	112	107	180
200						326	303	294	274	262	256	242	234	226	214	202	193	185	178	171	163	156	147	142	136	131	125	119	200

Square Feet of Radiation	Section 4																												Square Feet of Radiation
	Height Above Midpoint of Heater in Feet																												
	4½–5	5–6	6–7	7–8	8–9	9–10	10–11	11–12	12–13	13–14	14–15	15–16	16–17	17–18	18–20	20–22	22–24	24–26	26–28	28–30	30–33	33–36	36–40	40–43	43–46	46–50	50–55	55–60	
10	34	30	27	24	23	22	21	20	20	19	18	18	17	16	15	14	14	13	13	12	11	11	10	10	10	9	9	8	10
15	52	45	41	37	34	32	31	30	30	29	27	26	25	24	23	22	20	20	19	18	17	16	15	15	14	14	13	12	15
20	77	65	58	52	48	45	42	41	40	38	37	35	34	33	31	29	27	26	25	24	22	22	21	20	19	18	17	16	20
25	95	81	71	64	59	55	52	49	50	48	46	44	42	41	38	36	34	33	31	30	28	27	26	25	24	23	22	21	25
30	130	108	93	83	76	70	66	62	60	57	55	53	51	49	46	43	41	39	38	36	33	33	31	30	29	27	26	25	30
35	156	125	108	97	88	82	77	72	70	67	64	61	59	57	53	50	48	46	44	42	39	38	36	35	33	32	30	29	35
40	200	155	132	116	105	97	90	85	80	77	73	70	67	65	61	58	55	52	50	48	45	43	41	39	38	36	35	33	40
45	226	175	149	131	119	109	101	96	90	86	82	79	76	73	69	65	62	59	56	54	50	49	46	44	43	41	39	37	45
50	220	170	150	135	124	115	107	100	100	96	91	88	84	82	76	72	68	65	63	60	56	54	51	49	47	45	43	41	50
60	266	216	187	167	152	141	131	124	120	115	110	105	101	98	92	86	82	78	75	72	67	65	62	59	57	54	52	49	60
70	310	251	218	195	178	165	153	145	141	134	128	123	118	114	108	101	96	91	88	84	78	76	72	69	66	63	60	57	70
80		325	282	246	220	201	187	174	161	153	146	140	135	130	122	115	110	104	100	96	90	87	82	79	76	73	69	66	80
90			300	234	226	207	194	183	181	172	165	158	152	147	137	129	123	117	113	108	101	98	92	89	86	82	78	74	90
100				301	271	250	230	216	201	191	183	175	169	163	153	144	137	130	125	120	112	108	102	99	95	91	86	83	100
110					300	274	254	234	221	210	206	193	185	180	168	158	150	143	138	132	123	119	113	109	104	100	95	91	110
120						300	278	260	241	230	219	210	202	196	183	173	164	157	150	144	134	130	124	118	114	109	104	99	120
130							300	280	261	249	238	228	219	212	198	187	178	170	162	156	146	141	133	128	123	118	112	107	130
140								301	281	268	256	245	236	228	214	202	191	183	175	167	157	152	144	138	133	127	121	115	140
150									301	287	274	263	253	245	229	216	205	196	187	181	168	163	154	148	142	136	130	124	150
160										303	292	280	270	261	244	230	218	209	200	193	179	174	164	158	152	145	138	132	160
180											329	315	304	294	274	259	246	235	225	217	202	195	185	177	171	163	155	148	180
200															305	288	273	261	250	240	224	217	205	197	190	182	173	165	200

TABLE 8—ARRANGEMENT OF DATA FOR DESIGN OF HOT WATER SYSTEM.

Radiator Number	Section of Line	Height of Mid-point of Radiator Above Mid-point of Heater	Square Feet for Radiator	Square Feet for Mains	Flow Riser (Table 1)	Return Riser (Table 1)	Flow Main (Table 2)	Return Main (Table 2)	Main Heater Flow Riser (Table 4)	Main Heater Return Riser (Table 4)	Orifices (Table 5)
1	2	3	4	5	6	7	8	9	10	11	12
11	4	8′-0″	160	160	1½	1½	2½	2½	.....	.....	¾
10	4	7′-4″	24	184	¾	¾	2½	2½	.....	.....	7/32
9	3	8′-0″	65	249	1	1¼	3	3	.....	.....	11/32
8	3	17′-0″	21	270	½	¾	3	3	.....	.....	11/32
7	3	17′-0″	50	320	1	1	3	3	.....	.....	5/32
6	2	17′-0″	40	360	¾	1	3	3	.....	.....	5/32
5	2	7′-6½″	51	411	1	1	3½	3½	.....	.....	5/32
4	2	16′-6½″	36	447	¾	1	3½	3½	.....	.....	11/32
3	2	17′-0″	21	468	½	¾	3½	3½	.....	.....	5/32
14	3	7′-6½″	93	93	1¼	1¼	2	2	.....	.....	11/32
13	2	17′-0″	99	192	1¼	1¼	2½	3	.....	.....	11/32
12	2	7′-6″	16	208	½	¾	2½	3	.....	.....	11/32
	Junction of Branches										
...	...	.....	.....	676	.....	.....	4	4	.....	.....	.....
2	1	17′-0″	50	726	1	1	4	5	.....	.....	5/32
1	1	8′-4½″	25	751	¾	¾	4	5	.....	3½	11/32
16	4	7′-8½″	125	125	1¼	1¼	.....	.....	.....	.....	11/16
15	1	7′-6½″	105	230	1¼	1¼	.....	.....	.....	2½	3/8
	Junction.....		.....	981	.....	.....	5	5	4	.....	.....
No.	3 &	No. 4....	57	.....	1	1	.....	.....	.....	.....	.....

ft., find that a 2½ in. flow main and a 2½ in. return main are required. For 184 sq. ft. a 2½ in. flow main and a 2½ in. return main are needed and so on. The short line at the bottom of the figure is 6 ft. + 24 ft. + 3 ft. + 7 ft., or 40 ft. long. Accordingly, use the column headed "30-40" to find the sizes of its mains.

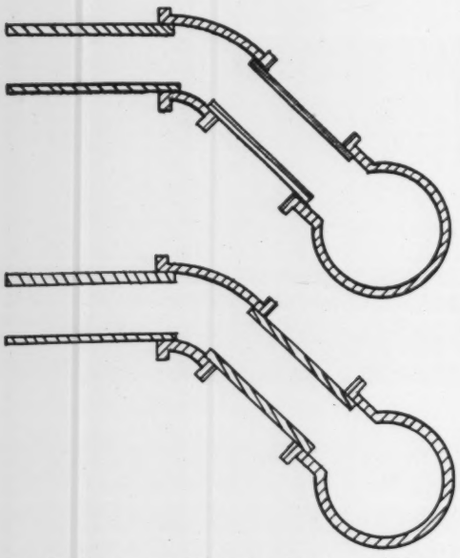

FIG. 7—SUGGESTED VERTICAL SECTION OF MAINS AND RISER LATERALS FOR LINES HAVING BRANCH MAINS

The numbers in column 10 are the sizes of the main heater risers. They are obtained by reference to Table 4 as previously. The numbers in column 11 are orifice sizes. Before these can be selected, the line must be divided into four equal sections. Since the longest branch is 72¼ ft. long, the sections will each be 18 ft. 1 in. long. This distributes the radiators among the sections as follows:

Section 1 contains radiators 1 and 2.
Section 2 contains radiators 3, 4, 5, 6, 12 and 13.
Section 3 contains radiators 7, 8, 9, and 14.
Section 4 contains radiators 10 and 11.

In the case of the small line at the bottom of the page:

Radiator 15 is in Section 1.
Radiator 16 is in Section 4.

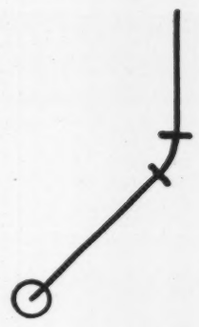

FIG. 9—LOW FRICTION CONNECTIONS FOR RISER LATERAL REQUIRING A BEND

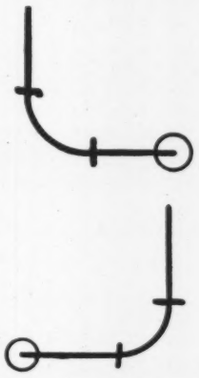

FIG. 10—ALTERNATIVE LOW FRICTION CONNECTIONS FOR RISER LATERAL REQUIRING A BEND

As previously noted, the section for each radiator is recorded in column 2 of Table 8. All radiators on the first floor except No. 1 come within the range of 7-8 feet above the heater; consequently their orifice sizes are found under the heading "7-8" in the proper sections of Table 5. The orifice size for No. 1 is found in Table 5, Section 1 under the column headed "8-9."

If it had been decided to use orifices calibrated for vapor systems or adjustable valves, the sizes or set-

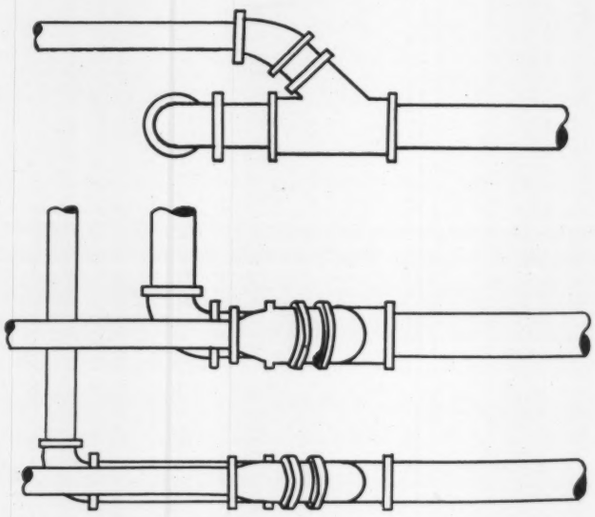

FIG. 8—METHOD OF TAKING A SHORT BRANCH OFF THE TOP OF A LONG BRANCH

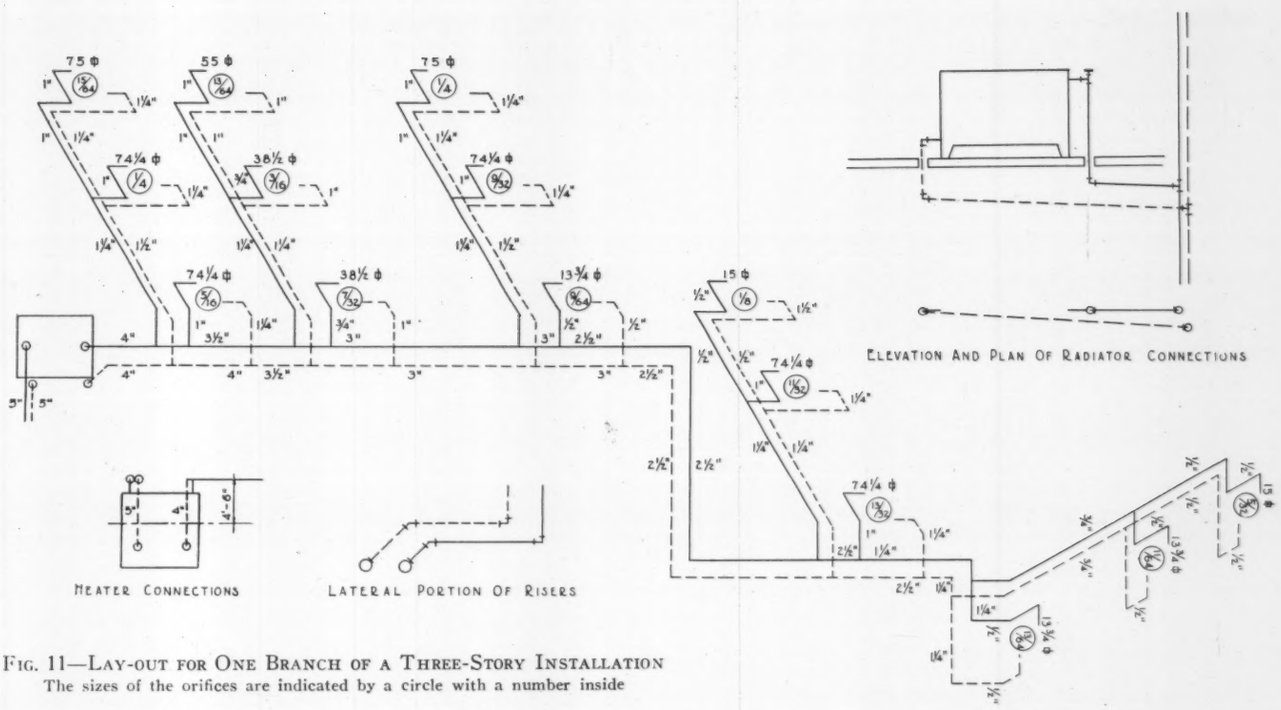

FIG. 11—LAY-OUT FOR ONE BRANCH OF A THREE-STORY INSTALLATION
The sizes of the orifices are indicated by a circle with a number inside

tings would have been as indicated in Table 9. There are two features of Table 9 that require comment. It will probably be noticed that the valve settings are always equal to or larger than the largest number in the corresponding vapor orifice rating. There are two reasons for this. First, the orifices, as nearly as could be determined by measurement, were correct at the upper range —that is, the 41-50 was actually 50, the 51-60 was actually 60, and so on. Thus, in comparing a valve setting to an orifice rating, attention should be given to the larger number in the orifice rating. Second, an orifice that is a little too small does no particular harm, while one that is too large may, to some extent, allow the riser pressure head to interfere with circulation. The orifice, therefore, is never too large, but may be as much as ten units too small. By taking both of the above considerations into account, it will be noted that the valve setting corresponding to a given orifice will nearly always be equal to or larger than the largest number on the orifice, but that it can never exceed it by more than 10. The second feature that requires comment is the limited range of the valve. None of them, so far as has been ascertained, can be set higher than 200. A certain amount of overloading will not do any harm and it is quite possible that these valves might be used successfully on radiators requiring a setting as high as 300, but on the other hand, the writer would hesitate to use them on a radiator requiring a setting of less than 30. When small settings are used the orifice in the valve becomes a narrow slit, and, therefore, might quite probably become clogged.

However, it is probable that the smaller settings recommended for these valves are too small because of the elongated shape of the orifice. There has as yet been no opportunity to test this experimentally, but since small readjustments can readily be made while the system is in operation, it should be easy to make satisfactory installations using these valves.

Systems for Buildings More than Two Stories High

It will be noticed that in the two-story system just considered all the first-floor radiators were independently connected to the mains. This was not only convenient, but necessary. When using these tables every radiator less than 12 ft. above the mid-point of the heater must be connected to the mains by independent risers. If this is not done the lower radiators will not heat prop-

TABLE 9—SIZES OF SETTINGS FOR ORIFICES CALIBRATED FOR VAPOR SYSTEMS OR ADJUSTABLE VALVES.

Radiator	Section of Line	Height Above Mid-point of Heater	Sq. Ft. of Radiation	Vapor Rating of Orifice	Valve Setting
11	4	8 ft.-0 in.	160	¾ in.	Impossible — must use 2, or better, 3 smaller radiators instead of this large one
10	4	7 ft.-4 in.	24	51-60	62
9	3	8 ft.-0 in.	65	101-110	118
8	3	17 ft.-0 in.	21	11-20	24
7	3	17 ft.-0 in.	50	41-50	59
6	2	17 ft.-0 in.	40	21-30	35
5	2	7 ft.-6½ in.	51	51-60	69
4	2	16 ft.-6½ in.	36	21-30	32
3	2	17 ft.-0 in.	21	0-10	19
14	3	7 ft.-6½ in.	93	151-160	169
13	2	17 ft.-0 in.	99	71-80	88
12	2	7 ft. 6 in.	16	11-20	21
2	1	17 ft.-0 in.	50	21-30	35
	1	8 ft.-4½ in.	25	11-20	24
16	4	7 ft.-8½ in.	125	11-16	Probably would be more satisfactory to use 2 smaller radiators
15	1	7 ft.-6½ in.	105	101-110	110

erly. However, for radiators more than 12 ft. above the mid-point of the heater, any number may be connected to the same riser, provided the capacity of the riser is not exceeded.

Fig. 11 shows the layout for one branch of a 3-story installation. For convenience the radiators have been numbered 11, 12, 13, 14, 15, on the first floor; 21, 22, 23, 24, 25, on the second floor; and 31, 32, 33, 34, 35, on the third floor. Thus the first digit represents the floor and the second the riser. The line is 97 ft. long and therefore each of the four equal sections into which the line must be divided are 24¼ ft. long. The distance from the mid-point of the heater to the mid-point of the main directly above is 4 ft. 4 in. The calculations are made exactly the same as for the previous cases except that for the second floor radiators the radiator connections and risers must be considered separately. The data for this installation are given in Table 10.

Similar tables can be made for either the vapor rated orifices or adjustable valves. It will probably be noticed that the main risers frequently do not need to be as large as the mains. Except for the increased cost there is no objection to enlarging them. Both the appearance and the circulation will be improved.

How to Improve the Circulation in Old Systems That Do Not Heat Properly

It is always desirable, of course, to see the system in actual operation, if possible. It should be carefully checked to make certain that there are no air pockets or other defects that might prevent circulation. Next, by means of Table 2 or Table 3, whichever is applicable, check the capacity of the main to see if it is as great or nearly as great as it should be. If the mains are large enough, and they usually are, the system should be supplied with orifices according to Table 5. If the first-floor radiators have independent risers this will remedy the trouble. If the first-floor radiators do not have independent risers, it may remedy the trouble anyway, but if it does not, a separate return main for these radiators will make them circulate and may be a cheaper remedy than the separate risers called for by the table. When the mains are too small probably the best solution is to run a separate line to relieve the overloaded mains. The one thing that should always be done is to put an orifice of the size indicated by Table 5 in each radiator.

Covering the Mains

These tables are based on the assumption that the mains are to be bare. Bare mains, particularly bare return mains, help circulation by increasing the heater temperature differential without any corresponding increase in the radiator temperature differentials. This increase in the heater temperature differential causes an increase in its pressure head and thereby helps circulation. It has been found by experiment that better results are usually obtained when the half of the flow main farthest from the heater is covered because the radiators near the end of the line then receive hotter water. For the same reason it has been found desirable to cover the smaller flow risers and flow laterals; ½-in. and ¾-in. flow risers and laterals give much better results when covered. Risers 1 in. or larger are not much affected by cooling, at least in installations of three stories or less. The return main should aways be left bare throughout its entire length.

TABLE 10—SUMMARY OF DATA FOR ONE BRANCH OF A THREE-STORY INSTALLATION.

Radiator	Section of Line	Height of Mid-point of Heater above Mid-point of Radiator	Square Feet for Radiator Connections	Square Feet for Risers Carrying more than one Radiator	Square Feet for Mains	Flow Radiator Connections (Table 1)	Return Radiator Connection (Table 1)	Flow Riser (Table 1)	Return Riser (Table 1)	Flow Main (Table 3)	Return Main (Table 3)	Main Heater Flow Riser	Main Heater Return Riser	Orifices
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
35	4	25′-10″	15			½	½							5/32
25	4	16′-11″	13¾	28¾		½	½	¾	¾					11/32
15	4	8′-1″	13¾			½	½							11/32
					42½					1¼	1¼			
34	3	25′-10″	15			½	½							1/8
24	3	16′-11″	74¼	89¼		1	1¼	1¼	1¼					11/32
14	3	8′-1″	74¼			1	1¼							11/32
					206					2½	2½			
33	2	25′-10″	75			1	1¼							1/4
23	2	16′-11″	74¼	149½		1	1¼	1¼	1½					5/32
13	2	8′-1″	13¾			½	½							5/32
					369¼					3	3			
32	1	25′-10″	55			1	1							11/32
22	1	16′-11″	38½	93½		¾	1	1¼	1¼					3/16
12	1	8′-1″	38½			¾	1	1¼						7/32
					501¼					3½	3½			
31	1	25′-10″	75			1	1¼							11/32
21	1	16′-11″	74¼	149¼		1	1¼	1¼	1½					1/4
11	1	8′-1″	74¼		724¾	1	1¼			4	4	3½[a]	3½[a]	11/32

a May be replaced by 4 in. risers if so desired.

If it is desired to construct a system having an unheated basement making it necessary to cover both pipes, the same tables can be used, provided the line is divided into three equal parts instead of four. The fourth section is the one to be eliminated. The effect of this change is to put increased friction into the radiator circuits which tends to prevent them from interfering with each other and thereby enables the reduced pressure head of the heater to keep the water moving through the mains as it should.

Systems Having Sealed Expansion Tanks

Although these tables were designed primarily for systems having open expansion tanks, they will work equally well for systems having sealed expansion tanks. If it is desired to design a system that will run on a lower temperature differential, this can be done by multiplying the number of square feet for each radiator by some constant greater than one, such as 1.5, before beginning to make a tabulation of the data.

Precautions to Be Observed

1. Be sure the mains are large enough, as straight as possible, and installed carefully enough to preclude the possibility of air pockets.

2. Be sure that all radiators less than 12 ft. above the midpoint of the heater are connected to the mains by independent risers.

3. Make the lateral portions of the risers as straight as possible.

4. Do not bunch too much radiation in the fourth section of the line. See that the first and second sections get a fair share of the radiation, even if it is necessary to make these two sections a trifle longer than they should be. When a riser comes on the border line be-

tween the two sections, it should always be placed in the section nearer the heater.

5. Some contractors make a practice of increasing the size of radiators at the end of the line farthest from the heater in order to compensate for poor circulation. When using these tables that should not be done, because every radiator can be counted upon to deliver about 150 B.tu per hour per square foot when the water is leaving the heater at 200 F. If the sizes of some of the radiators are increased, the house will not be uniformly heated.

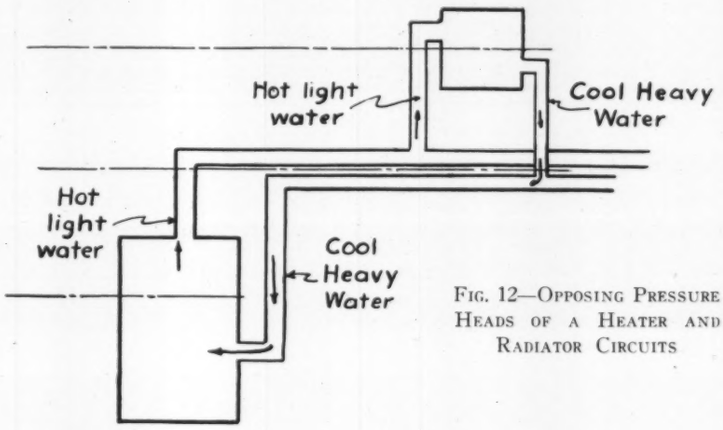

FIG. 12—OPPOSING PRESSURE HEADS OF A HEATER AND RADIATOR CIRCUITS

6. Finally, be sure to put an orifice of the proper size in every radiator. The omission of a single orifice may be sufficient to cause a great deal of trouble.

7. It is a good plan to cover all ½ in. and ¾ in. flow laterals and risers as well as about 1/3 or ½ of the flow main beginning at the end farthest from the heater.

How the Tables Were Calculated

In making the tables a special effort was made to accomplish the following things:

1. To make sure that the radiators at the ends of the lines would get as hot as those near the heater, no matter how long the lines were.

2. To make sure that the radiators on the lower floors would get as hot as the radiators on the upper floors.

3. To provide a safety factor so large that the circulation would be very good, even though the system was somewhat overloaded as, for instance, by adding a radiator or two after the installation had been completed.

4. To make the tables as simple as possible, especially those portions that relate to pipe sizes.

It has been impossible to test the tables experimentally through their entire range, but the tests that have been made have been so very satisfactory that the first three aims have been very largely realized. It is hoped that the fourth will also be accomplished. Attempts to produce simple tables have cost more labor than any other factor.

Why Radiators at the Ends of Lines Frequently Do Not Heat Well

In order to make clear the method of producing strong circulation through the radiators on the ends of the lines, it will be necessary to refer briefly to a paper entitled "Pipe Sizes for Hot Water Heating Systems" by Giesecke and Smith published in the A. S. H. V. E. JOURNAL, June, 1929. In this paper it was pointed out that the principal cause of cold radiators on the end of a line was the high pressure built up in the return main by the radiators nearer the heater. This fact is so important that to ignore it is to invite almost certain failure in certain parts of the system. Just how it affects the calculation of pipe sizes may be seen from the following considerations.

When making calculations on hot water heating systems it seems to have been customary to assume that the flow through the mains was caused by the difference in weight of two columns of water extending from the middle of the heater to the middle of the radiator. For underfoot distribution systems this particular assumption cannot be justified and has undoubtedly been one of the causes of disappointing performance. In order to make a radiator start in the right direction it is necessary that the pressure in the flow main at the point of junction with the flow riser be higher than the pressure at the corresponding point in the return main. If the pressure in the return main is higher the radiator will start backward. It will also be sluggish and never really hot. It is very important then to keep the pressure in the flow main higher than the pressure in the return main. Fig. 12 shows that the pressure head of the heater does tend to raise the pressure in the flow main. The cool, heavy water in the return heater riser tends to force water into the bottom of the heater and out of the top, thus producing the desired higher pressure in the flow main.

As to the radiator, the cool, heavy water is flowing directly into the return main, thereby tending to increase the pressure in exactly the wrong place. The conclusion is, therefore, that except for its own particular circuit, the pressure head generated in a radiator and its risers, instead of helping circulation, is a great disturbing influence. The problem of insuring good circulation is, then, two-fold.

1. The resistance of the mains must be low enough

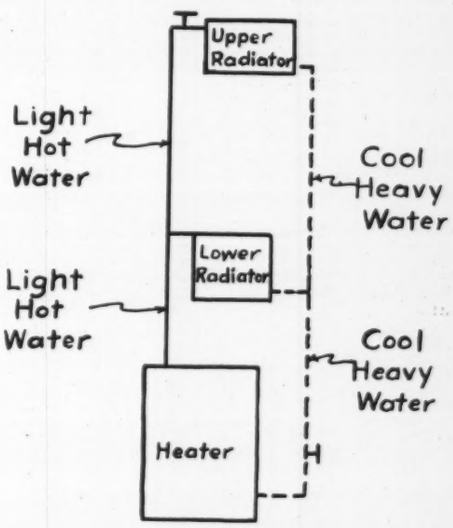

FIG. 13—OPPOSING PRESSURE HEADS IN A RISER

so that the heater pressure head alone is great enough not only to overcome all the friction in the mains, but also to leave a surplus large enough to insure that throughout the entire length of every line the pressure in the flow main shall be higher than the corresponding pressure in the return main.

2. The friction head in every radiator circuit must be great enough not only to absorb all its own pressure head, but to absorb, in addition a pressure head equal to the difference in pressure between the mains at the points where its risers join the mains.

Why Radiators on Lower Levels Frequently Do Not Function Well When Connected to the Same Riser with High Radiators

Having located the trouble with radiators at the ends of lines, consider some of the difficulties that beset first-floor radiators. In order to make the explanation as clear as possible, note that the line in Fig. 13 has two valves in it, one near the top and the other near the bottom. Under normal conditions the flow riser on the left will be full of light, hot water and the return riser on the right is full of cool, heavy water. Now, if the upper valve is closed, it is evident that the cool, heavy water column in the return riser will displace the column of light, hot water in the flow riser and the radiator will circulate strongly in the right direction. Suppose the upper valve be opened and the lower one closed. The cool, heavy column of water in the return riser now tends to flow backward through the radiator, just the opposite of its former action. From this it is apparent that the two columns of water below a radiator tend to make it flow in the right direction, while the columns above it tend to make it flow in the wrong direction. It is evident that if the lower radiator is to circulate properly, the head generated by the water columns above it must be neutralized by adding friction to the upper radiator. Similarly, the friction in the lower pipes should be as small as possible.

Unfortunately, in any actual circuit there are necessarily a number of bends just below the floor. In order to remedy this situation it was decided to choke the higher radiators by means of orifices, but calculations showed that this alone was not sufficient. There were two other things that could be done:

1. Make the lower portions of the risers very large.
2. Avoid placing radiators in the danger zone, but connect all radiators below a certain safe level to the mains by separate risers.

From the standpoint of calculations the first alternative was more simple, but from the standpoint of the tables required it was complicated. Also, it seemed probable that the second method would give a better appearing installation.

It will be noticed that the problem of the low radiator is very similar to the end-of-the-line radiator. In each case a pressure head from some other radiator or radiators is producing interference, and the remedy is twofold:

1. As far as practicable neutralize the conflicting head by friction.
2. Preserve the strength of the favorable pressure head as far as possible by means of low resistance, or, if this for some reason is undesirable, so arrange the system that no radiators will be connected between points where the pressure is negative.

As far as the mains were concerned, little difficulty was experienced. It was assumed that the heater, its main risers and fittings, including the turns at the tops of the main risers, would absorb 20 per cent of the available pressure head. The capacities given in Table 4 were thus easily selected. It was further assumed that 30 per cent of the heater pressure head was to be available at the ends of the lines. This left 50 per cent to be dissipated in friction in the mains. Accordingly, it was assumed that the mains might have four turns in addition to those already accounted for in the heater connections and the values were calculated for Tables 2 and 3 accordingly.

Since it was considered desirable to leave the mains bare in order to heat the basement, it was inevitable that the water should be somewhat cooled before reaching the farthest radiators. In order to compensate for this it was decided to run the radiators in the different sections of the line at various temperature differentials. The radiators in Section 4 operated on a 25-F differential; those in Section 3 on a 30-F differential; those in Section 2 on a 35-F differential, and those in Section 1 on a 40-F differential. This makes the average temperature of all the radiators about the same and they should all deliver about 154 B.tu per hour per square foot of radiation when the water is leaving the heater at 200 F.

The risers all have a friction head of about 16 mill inches per foot, including both pipes. This applies only to the vertical portions. The crooked portions in the basement have a somewhat high and irregularly varying resistance. This makes the pressure to be absorbed by the orifices also an irregular variable so that the capacities cannot be checked by plotting a curve, except for the higher radiators.

It was through the kindness of Prof. C. W. Crawford, head of the Department of Mechanical Engineering and Prof. M. V. Brewer of the same department that it was possible to find a place to install the three story experimental plant. In a recent test on this plant with one-half the flow main and the smaller flow risers covered, the hottest radiator in the system was only 8½ F hotter than the coldest. The average temperature was a trifle over 170 F.

Suggested Method of Testing Unit Heaters Suitable for Field Use

By L. S. O'Bannon[1], Lexington, Ky.
MEMBER

IN TESTING unit heaters it has been the experience of most observers that the warm air leaving the unit does not have a uniform temperature or velocity. The average of the readings of several thermometers located at the discharge outlets *may not* give the true average temperature of the total weight of air discharged. Instances have been reported, however, where satisfactory temperature readings have been made in this way but these instances seem to be exceptions. Accuracy in observing the final temperature of the air is vital to the condensate-temperature rise method of computing air volume. If this method is used it is necessary either to obtain a weighted average, which requires the determination of velocities, or to force a uniform temperature by providing some means of mixing the air. The latter system is generally favored. The weighted average method is seldom attempted.

It is the purpose of the writer to describe a method by which the air volume output may be obtained accurately with a minimum amount of auxiliary equipment. The method may be used in the laboratory or in the field. It is especially recommended for field tests, first, because of its simplicity, and second, because its simplicity is not secured by sacrificing accuracy.

The method has been named the condensate-nozzle method. The underlying principles are the same as those upon which the condensate-temperature rise method is based. The heat output of the unit is found in the usual manner by measuring the steam condensing capacity of the heater. Air volume outputs are computed indirectly from the temperature rise and the heat given up by the steam. The distinctive feature of the condensate-nozzle method is the graphical means employed to arrive at certain unobserved final temperatures. The procedure is as follows:

Let it be required to obtain for three different fan speeds, say 1750, 1160 and 870 r.p.m., the heat output, air volume output and final air temperature of a unit heater of the draw-through type illustrated in Fig. 1. The steam pressure is to be 5 lb. per square inch. The unit is arranged for a heat output test as shown, and, it should be noted, is free from any abnormal restrictions to air flow.

Three heat output tests are made, as near the specified speeds as possible. Let it be assumed that these tests yield the data as given in Table 1, Section A. No observations of final temperature were made and therefore it is impossible to calculate the air volume output.

The next step is to remove the diffusing outlets of the unit and place in their stead a venturi-shaped nozzle as shown in Fig. 2. The inlet section of the nozzle covers all the outlets of the unit. The throat has a relatively small diameter sufficient to cause an air velocity of 1,000 f.p.m. or more as determined from the lowest air volume

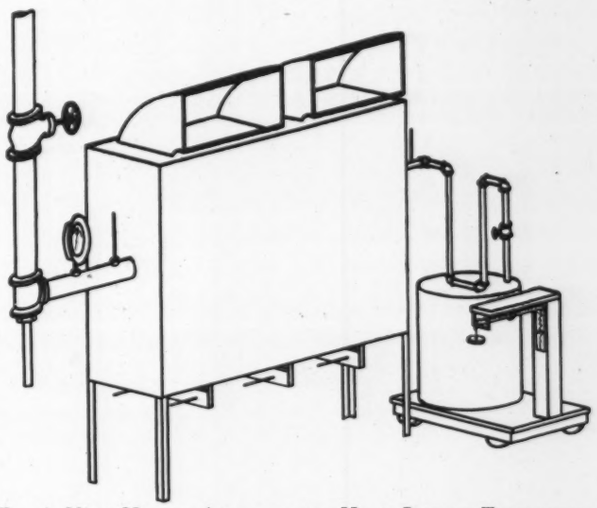

FIG. 1. UNIT HEATER ARRANGED FOR HEAT OUTPUT TEST WITHOUT ANY ABNORMAL RESTRICTIONS TO AIR FLOW

output anticipated. The purpose of the diverging outlet cone is to reduce the over all resistance of the nozzle. Four thermometers are inserted at the throat section for measuring the final temperature. Three more tests are now made with results as shown in Table 1, Section B.

If the data of tests 4, 5 and 6 are used to plot a curve showing the relation between final temperature and heat output this curve can be referred to, to obtain the final temperatures for tests 1, 2 and 3. However, the relation between heat output and final temperature varies with the entering air temperature. Therefore it is necessary to reduce the data of tests 4, 5, and 6 to the same entering air temperature as tests 1, 2 and 3. This requires three separate transformations since the values of the entering air temperature for the first three tests are not the same. Leaving the method by which the transformations are made for later discussion, Table 1, Section C, gives the corrected values.

The curves are plotted as shown in Fig. 3. The final temperatures corresponding to the heat outputs of tests 1, 2 and 3 are read from the curves, thus completing the data of these tests as shown in Table 1, Section D. Since an entering air temperature of 60 F is used as a basic temperature for the purpose of rating, the transformation to this base may be made most conveniently at this

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stage before continuing the calculations. The converted figures are given in Table 1, Section E.

The air volume outputs are now calculated. The curves are plotted as shown in Fig. 4, giving the relations between fan speed, the volume of air in cubic feet per minute, the final temperature and the heat output in B.tu per hour. It only remains to select from these curves the proper values corresponding to the three fan speeds specified at the beginning. The results are summarized in Table 1, Section F.

If it had been required to find the output of the unit at only one speed, for example, at the rated speed of an alternating current motor, the three points required for the curve in Fig. 3 could have been obtained just as readily by dampering the air flow at the outlet of the nozzle or by inserting orifices at the throat.

Although the diverging cone of the nozzle helps to reduce the over-all resistance and minimizes the hazard of extrapolating to get the final temperature at the maximum output, this cone is not essential. If there is some means available for increasing the normal speed of the fan, it is possible to exceed the normal maximum output, eliminating extrapolation entirely and permitting the use of a nozzle of higher resistance. A substitute motor of higher speed may be used; a d-c motor may be run for the short period of the test at a few hundred revolutions above its normal speed; an auxiliary motor with belt drive may be used.

It should be noted especially that this method is not dependent upon the accurate reproduction of free delivery conditions within the confines of the duct placed over the outlets. The method does not prohibit the use of an auxiliary fan to overcome the resistance of the collecting nozzle or to exhaust the heated air out of the test room, but the method emphasizes the fact that the auxiliary fan is not essential from the standpoint of totally equalizing or balancing the resistance of the nozzle; furthermore, the method is not concerned with the volumetric proportions of the nozzle, or other mixing device which may be substituted for it.

TABLE 1. SUMMARY OF TEST DATA

Section	Test No.	Fan Speed R. P. M.	Inlet Air Temp.—F.	Outlet Air Temp.—F.	Heat Output B. T. U. per hour	Air Volume cu. ft. per min. at Fans
A	1	1751	87.9	.....	182,900	....
	2	1184	87.8	.....	135,500	....
	3	788	90.7	.....	97,600	.....
B	4	1752	68.9	133.0	205,300	....
	5	1175	69.7	139.8	153,400	....
	6	794	74.1	149.2	109,600	....
C	4	1752	87.9	145.1	179,600	....
	5	1175	87.9	150.6	134,800	....
	6	794	87.9	156.7	99,200	....
	4	1752	87.8	145.0	179,600	....
	5	1175	87.8	150.5	134,800	....
	6	794	87.8	156.6	99,200	....
	4	1752	90.7	146.8	175,600	....
	5	1175	90.7	152.3	132,000	....
	6	794	90.7	158.3	97,200	....
D	1	1751	87.9	144.9	182,900	....
	2	1184	87.8	150.5	135,500	....
	3	788	90.7	158.2	97,600	....
E	1	1751	60	126.8	221,000	3403
	2	1184	60	133.8	164,000	2310
	3	788	60	141.3	120,900	1567
F		1750	60	126.8	221,000	3390
		1160	60	134.2	162,000	2280
		870	60	139.7	130,000	1720

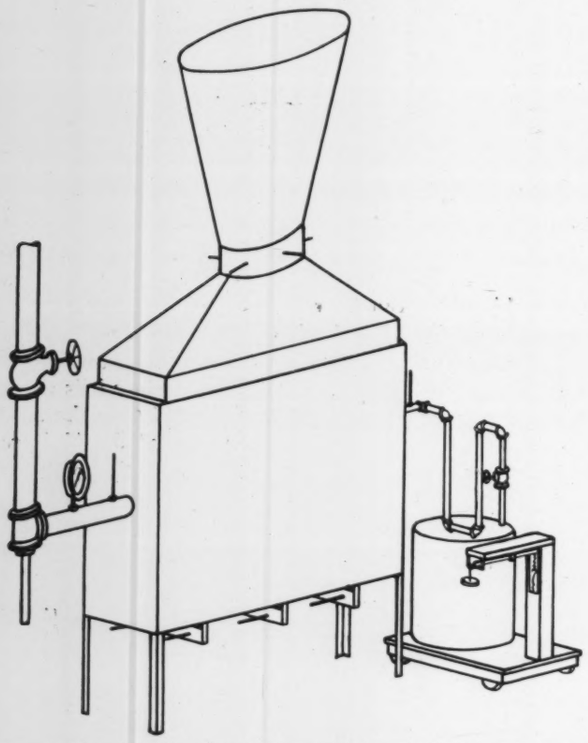

FIG. 2. UNIT HEATER WITH VENTURI NOZZLE ATTACHMENT FOR AIR VOLUME TEST

Attention has been called to the suitability of this method for field tests. While there may not be much demand for field tests of unit heaters there may be installations where ventilation is of prime importance; and outside air connections, mixing dampers and filters may impose frictional resistance materially affecting the air volume. In such instances it may be desirable to have a relatively simple and inexpensive method by which the air volume output may be accurately determined. Even in cases of this kind, a field test designed to determine air volume especially is not necessary if the unit in question has been tested according to the rules of the *Industrial Unit Heater Association*. Because, as the foregoing discussion bears out, it is only necessary to measure the steam condensing capacity of the unit and the entering air temperature in order to obtain the air volume if there is already available reliable information in the form of curves or catalogue data showing the relation between air volume and heat output. The majority of catalogues furnish these data. That is to say, if a unit heater is operating under abnormal restrictions to air flow, either at inlet or at outlet or both, the air volume output may be obtained by measuring the condensation rate of the unit and referring the heat output

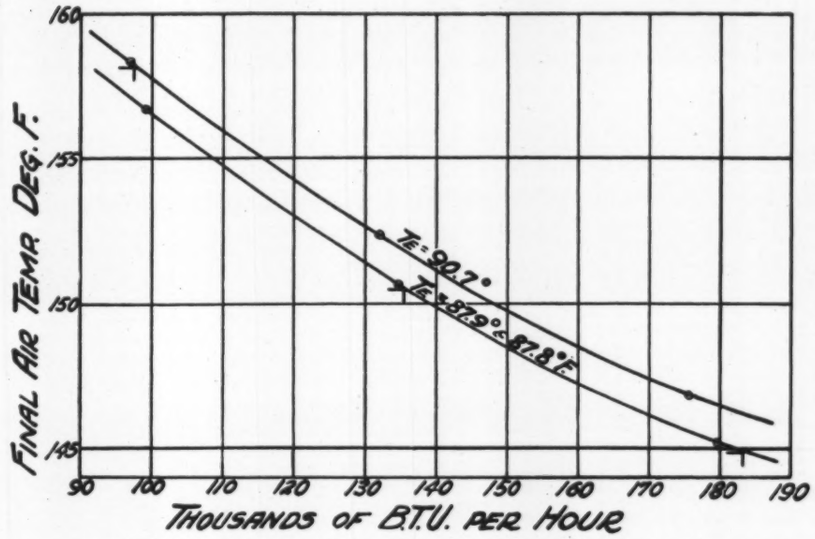

FIG. 3. RELATION BETWEEN FINAL AIR TEMPERATURE AND HEAT OUTPUT

to a curve derived from the manufacturer's catalogue data.

It would seem, therefore, of little use to have a method simple enough and inexpensive enough to take to the field. On the other hand, the availability of such a method may be welcomed by contractors, purchasers and consulting engineers. The method is also suitable for the occasional testing which may be undertaken in private laboratories or institutions where the extent of the work would not justify more elaborate equipment.

Contrasting this method with the direct measurement of air volume by Pitot tube or other flow meter and with the method of artificial reproduction of free delivery, the advantages are:

1. Small amount of test equipment required.
2. Minimum amount of test labor required.
3. Minimum floor space required.
4. Relatively inexpensive, quickly set up, and readily adapted to the occasional test in laboratory or field as contrasted with the more elaborate and permanent set-up probably more convenient for rapid routine testing.
5. The air volume output, although obtained indirectly, is the volume produced without abnormal restrictions to air flow.

The disadvantages are:

1. No single test is complete in itself.
2. At least three extra tests are required regardless of the number of fan speeds specified for which data are wanted. This is time-consuming and is a serious objection where a large amount of routine testing is performed.
3. Except under favorable circumstances, inability to get rid of heated air in the test room.
4. Unless special attention is given to their design, individual nozzles are not adaptable to a range of unit heater sizes or shapes.

Fig. 5 illustrates other applications of the nozzle. On account of the relatively small surface area of the nozzle between the heater and the throat section, insulation is not very important, but the loss should not be disregarded and plenty of insulation should be provided.

In the preceding explanation of the test method, the manner of converting the test data from one entering air temperature to another was not discussed. Several investigators have shown that the air volume at the fan of a unit heater remains constant for a given speed regardless of changes in inlet air temperature and steam temperature. This law can be used to derive formulae by which the final air temperature and heat ouput may be converted from one set of conditions to another. The derivation of these conversion formulae for a draw-through heater is as follows:

Let Q = Heat output, B.tu. per hour
W = Weight of air, pounds per hour
V_F = Volume of air at fan, cubic feet per min.
C_p = Specific heat of air
U = Coefficient of heat transmission
S = Heating surface
T_S = Steam temperature, degrees Fahrenheit, absolute
T_F = Final temperature of air, degrees Fahrenheit, absolute
T_E = Entering temperature of air, degrees Fahrenheit absolute
d = Density of air

Subscript—*1* refers to some known initial condition.
Subscript—*2* refers to a second condition to which the first is to be converted.

Then:

$$\frac{Q_1}{Q_2} = \frac{W_1 C_p \ (T_{F_1} - T_{E_1})}{W_2 \ C_p \ (T_{F_2} - T_{E_2})} \qquad (1)$$

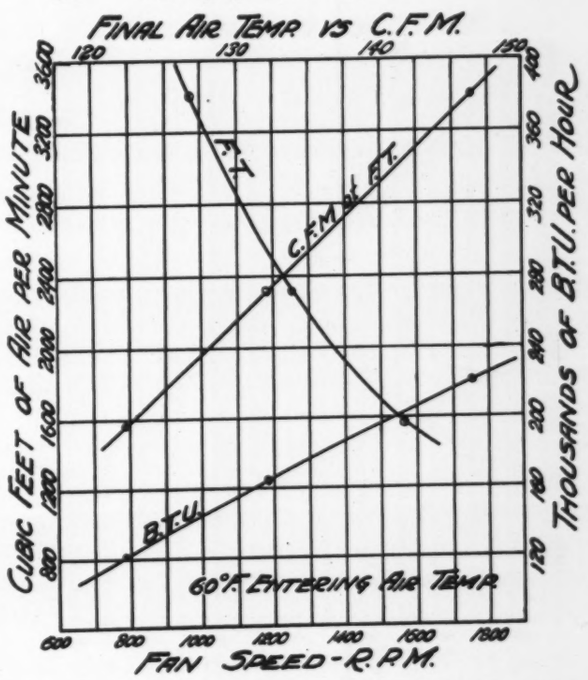

FIG. 4. CURVES SHOWING RELATION BETWEEN FAN SPEED, HEAT OUTPUT, AIR VOLUME AND FINAL TEMPERATURE

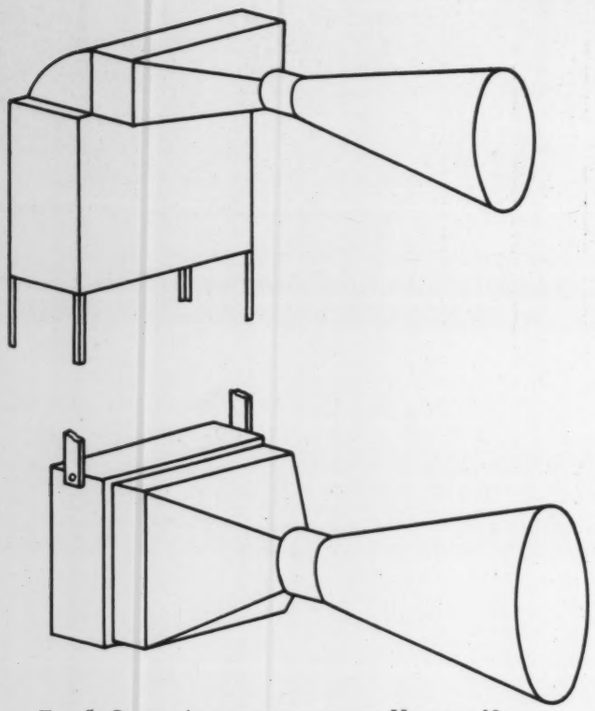

FIG. 5. OTHER APPLICATIONS OF THE VENTURI NOZZLE

$$\frac{W_1}{W_2} = \frac{60\ V_{F_1}\ d_1}{60\ V_{F_2}\ d_2} \tag{2}$$

$$\frac{d_1}{d_2} = \frac{T_{F_2}}{T_{F_1}} \tag{3}$$

Combining (2) and (3) with (1):

$$\frac{Q_1}{Q_2} = \frac{V_{F_1}\ T_{F_2}\ (T_{F_1} - T_{E_1})}{V_{F_2}\ T_{F_1}\ (T_{F_2} - T_{E_2})} \tag{4}$$

But the volume at the fan remains constant.
Therefore $V_{F_1} = V_{F_2}$, and

$$\frac{Q_1}{Q_2} = \frac{T_{F_2}\ (T_{F_1} - T_{E_1})}{T_{F_1}\ (T_{F_2} - T_{E_2})} \tag{5}$$

or

$$\frac{Q_1}{Q_2} = \frac{1 - \dfrac{T_{E_1}}{T_{F_1}}}{1 - \dfrac{T_{E_2}}{T_{F_2}}} \tag{6}$$

Also, from the familiar heat transmission formula, $Q = US\theta_m$, in which θ_m is the logarithmic mean temperature difference:

$$\frac{Q_1}{Q_2} = \frac{U_1 S\ (T_{F_1} - T_{E_1})}{\log_e\left(\dfrac{T_{S_1} - T_{E_1}}{T_{S_1} - T_{F_1}}\right)} \div \frac{U_2 S\ (T_{F_2} - T_{E_2})}{\log_e\left(\dfrac{T_{S_2} - T_{E_2}}{T_{S_2} - T_{F_2}}\right)} \tag{7}$$

Assuming $U_1 = U_2$ and equating (7) to (5) we have

$$\frac{T_{F_2}}{T_{F_1}} = \frac{\log_e\left(\dfrac{T_{S_2} - T_{E_2}}{T_{S_2} - T_{F_2}}\right)}{\log_e\left(\dfrac{T_{S_1} - T_{E_1}}{T_{S_1} - T_{F_1}}\right)} \tag{8}$$

Rearranging equation 8:

$$\frac{T_{F_1}}{\log_e\left(\dfrac{T_{S_1} - T_{E_1}}{T_{S_1} - T_{F_1}}\right)} = \frac{T_{F_2}}{\log_e\left(\dfrac{T_{S_2} - T_{E_2}}{T_{S_2} - T_{F_2}}\right)} \tag{9}$$

It is apparent therefore, that for any given heater and fan speed:

$$\frac{T_F}{\log_e\left(\dfrac{T_S - T_E}{T_S - T_F}\right)} \text{ is a constant.}$$

Making use of this relation, charts or tables can be constructed by means of which the new final temperature may be found corresponding to any new entering air temperature or steam temperature. After finding the new final temperature, equation (6) can be used to find the new heat output.

Similarly, for a blow-through unit heater, the following relation is obtained:

$$\frac{Q_1}{Q_2} = \frac{\dfrac{T_{F_1}}{T_{E_1}} - 1}{\dfrac{T_{F_2}}{T_{E_2}} - 1}$$

pue

$$\frac{T_E}{\log_e \dfrac{(T_S - T_E)}{(T_S - T_F)}} = \text{a constant.}$$

The transformations involved in the example of test calculations described in the foregoing were made by means of the function just derived for draw-through heaters. It would have been simpler but not quite correct if the conversions had been made according to the assumption of constant mass velocity using the familiar relations

$$\frac{Q_1}{Q_2} = \frac{T_{F_1} - T_{E_1}}{T_{F_2} - T_{E_2}} = \frac{T_{S_1} - T_{E_1}}{T_{S_2} - T_{E_2}}$$

Using the constant mass formula, all of the tests could have been converted immediately to equivalent conditions at an entering temperature of 60 F and the intermediate steps made necessary by the constant volume method would have been unnecessary.

Tests of Disc and Propeller Fans

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MEMBER

ALONG with the industrial growth of recent years there has developed a greater need for fans for man-cooling and for the removal of fumes and obnoxious gases in industrial plants; greater needs for the cooling of offices, restaurants and kitchens have also been realized. These needs have been filled largely by the increased application of fans of the disc and propeller types, and as a result of the increased demand a number of new designs of fans of these types have been developed. Prominent among the newer designs are the fans of the propeller types having blades similar in shape to the airplane propeller.

In many installations considerable saving in power consumption has been effected by the replacement of centrifugal fans by disc fans, and in other cases opposite results have been found, and many controversies have arisen as to the comparative performances of the various types and designs.

Reliable performance data for centrifugal fans are available but similar data relating to disc and propeller fans are limited, especially in the case of fans of the airplane propeller type.

Object

The purposes of this paper are two-fold: (1) to note and discuss some observations which the writer has made in the testing of disc and propeller fans, and (2) to present typical data and performance curves for fans of the airplane propeller type.

The observations herein noted are based on commercial tests of fifteen or more fans of various sizes and designs which have been conducted in the Mechanical Engineering Laboratory of the Ohio State University, and the data are presented on subsequent experimental work which is the outgrowth of these commercial tests, and which is now in progress as a project of the Engineering Experiment Station of the same University.

Fan Ratings

The most favorable field of service for disc and propeller fans is in moving relatively large quantities of air against small resistances, as in this field they are generally less costly than the centrifugal fans and occupy less space. Certain disc and propeller fans are designed especially for moving small quantities of air against higher resistances, but the majority are found in the former field of service and the manufacturers therefore usually rate their fans on the basis of their free air capacity in cubic feet per minute, and the accompanying speed in revolutions per minute and power input in watts or horsepower.

A little study of these ratings will show some interesting comparisons and some gross errors or exaggerated claims, based, no doubt, upon the results of inaccurate methods of air measurement, and in a few cases with no apparent basis other than that Manufacturer *A* believes his fan to be superior to that of Manufacturer *B* and therefore feels justified in publishing ratings which are somewhat more optimistic.

As an illustration of widely different claims for fans of similar design, the following is noted in the catalog data published by two manufacturers. The first, a large manufacturing concern with an extensive research department, rates their 36 in. fan at 10,700 c.f.m. when running at 945 r.p.m. with an input of 1 hp. The second, a smaller and less widely known concern, rates a smaller fan, their 24 in., at a displacement of 35,000 c.f.m. when running at 1,200 r.p.m. with the same input of 1 hp. If by displacement the manufacturer means the rated air capacity, the capacity of the latter fan is claimed to be 3.27 times that of the former. The first manufacturer further claims for his fan an efficiency of 31 per cent for free discharge. The second manufacturer publishes no values of efficiency but if he had done so on the basis of his published ratings he would have had to claim an efficiency for free discharge of over 3,000 per cent, as shown by the following analysis.

Power Required by a Perfect Fan

If a fan could be made to operate without any losses, such as those due to eddy currents, air friction and leakage, the power input would be equal to the power required to move the air against the existing pressure. The power required to move the air as expressed in horsepower is equal to the weight of air in pounds per minute, times the head in feet of air, divided by 33,000 ft.-lb. per minute. This expression when converted to more convenient terms states that the air horsepower equals the volume of air in cubic feet per minute times the total pressure in inches of water, divided by 6,356. (See A. S. H. V. E. Code for Testing Centrifugal and Disc Fans.) When a fan is discharging freely into the atmosphere no appreciable static pressure exists at the plane of the fan blades, and the total pressure therefore is equal to the velocity pressure, in inches of water, or

$$D\left[\frac{V}{1096.2}\right]^2$$

where V is the velocity in feet per minute and D is the density of the air in pounds per cubic foot. For average conditions of barometer and temperature the density of the air is approximately 0.072 lb. per cubic foot, and by substituting this value for D, the expression for velocity pressure becomes

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$\left[\frac{V}{4100}\right]^2$.

Applying this expression to the ratings cited for the 24-in. fan, 35,000 c.f.m. must pass through the opening of the fan, the area of which in this case is approximately 3.5 sq. ft., with a resulting average velocity of 10,000 f.p.m.; the velocity pressure, $\left[\frac{V}{4100}\right]^2$ is therefore $\left[\frac{10{,}000}{4100}\right]^2$ or 5.95 in. of water, and the power required to move the air is $\frac{35{,}000 \times 5.95}{6356}$ or 32.7 hp. No account has here been taken of the decrease in the opening of the fan due to the space occupied by the fan blades and shaft, but if such account were taken the horsepower requirement would be even greater than the calculated value of 32.7 hp. The manufacturer, in claiming to perform an amount of work equal to 32.7 hp. with a 1 hp. motor, is obviously claiming the impossible, and, whether unknowingly or by intent, is taking an unfair advantage of his competitors who have honestly and intelligently rated their fans.

If it were possible to make a fan to operate with 100 per cent efficiency, in the case cited the volume of air which could be passed through an opening of 3.5 sq. ft. by the expenditure of 1 hp. would be 10,920 c.f.m. or only 31 per cent of that claimed by the manufacturer. It should be noted that the air capacity claimed by the manufacturer exceeds that of a theoretically perfect fan in the ratio of 3.2 to 1, whereas the horsepower requirement exceeds that of the theoretically perfect fan in the ratio of 32.7 to 1. The horsepower varies as the cube of the fan capacity, and therefore an analysis of performance on the basis of the horsepower required to move the air will magnify discrepancies in ratings that are not so apparent when comparisons are made directly on the volume (c.f.m.) basis.

Table 1 shows the free-discharge capacities of fans of various diameters that would be possible if they could operate without air friction or leakage or any other losses. If a fan actually delivers 75 or 80 per cent of the tabulated value, its performance should be considered excellent.

TABLE 1. FREE-DISCHARGE CAPACITY OF FANS IN CUBIC FEET PER MINUTE AT 100 PER CENT EFFICIENCY

Horsepower Input to Fan Blades	Diameter of Fan Opening or Outlet, Inches				
	12	18	24	30	36
0.25	2550	4370	6420	8640	11000
0.50	3210	5500	8100	10900	13860
0.75	3670	6290	9250	12440	15830
1.00	4040	6920	10180	13690	17400
1.50	4620	7930	11650	15660	19950
2.00	5100	8740	12840	17280	22000

In comparing the rating of any fan with the values of Table 1 it should be noted that some manufacturers catalog their fans upon the diameter of the blade circle, others upon the diameter of the fan opening, and still others upon the overall dimensions. For any values of the diameter of the fan opening other than those shown in the table, at equal power inputs the capacity varies as the $\frac{4}{3}$ power of the diameter ratio; and for a fixed diameter of fan opening the capacity varies as the cube root of the ratio of power inputs.

Methods of Testing

The testing of disc and propeller fans is somewhat more difficult than the testing of centrifugal fans owing to the lack of a satisfactory means of measuring the velocity of a swirling stream of air.

A common but unsatisfactory method of testing consists of traversing the stream of air on the discharge side of the fan with an anemometer or a Pitot tube. Results obtained by this method are inaccurate not only because of the error in measuring a swirling flow but also because of the difficulty in measuring the cross-sectional area of the stream of air which may be greater or less than the area of the opening of the fan and furthermore varying at different distances from the fan. The velocity also varies in different parts of the stream, and with some designs of fans the air near the fan blades may even be flowing in opposite directions, although this fact might not be detected by an anemometer; it can, however, readily be detected by holding a thread in or near the stream and observing the direction which it takes, or by a smoke test. Where a fan which is intended to exhaust air from a room is continually obtaining a part of its supply by leakage from the discharge side, outside of the room, a measurement of the stream on the discharge side is not a true indication of the useful capacity of the fan.

In order to overcome some of these difficulties in air measurement, some fan testers have employed the method of arranging a short length of metal duct on the discharge side of the fan so as to confine the stream of air within an area of known cross-section. This method undoubtedly produces more consistent results than that of attempting to measure the unconfined stream. It is still, however, more or less subject to the errors due to the swirling effect unless this swirl is straightened out by the use of straightening vanes in the duct, and these vanes offer additional resistance of unknown value. The duct may also very greatly affect the capacity of the fan by converting velocity pressure into static pressure and by reducing eddy currents, as will be shown later.

Standard Code Method

The Standard Code for the Testing of Centrifugal and Disc Fans which was adopted in 1923 by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers* specifies a method of testing disc and propeller fans which is quite different from the methods just

described. This method is fundamentally that of measuring the quantity of air which is discharged by a centrifugal fan into an air chamber maintained at zero pressure when that quantity is therefore just sufficient to replace that which is being exhausted from the chamber by the disc or propeller fan. Under this method the air flow is measured by a Pitot tube which is located on the discharge side of the centrifugal fan where little or no swirl exists and where the stream of air can be measured with accuracy. A diffuser is required in the air chamber so as to guard against short-circuiting of air from the discharge pipe of the centrifugal fan, through the chamber, to the disc or propeller fan. A draft gage, connected to an impact tube inserted into the chamber, is used to indicate the pressure in the chamber.

This method involves the set-up of a considerable amount of apparatus and for that reason has not been used as generally as some of the more simple but less satisfactory methods. Without doubt, if purchasers of disc and propeller fans demanded fan ratings according to the A. S. H. V. E. Standard Code method, and if manufacturers generally adopted this method of testing, there would be fewer cases of such unfair and often absurd claims as are now frequently found.

Comparison of Ratings by Different Test Methods

No attempt is here made to compare ratings by the Standard Code method with those by the less accurate methods in that the results by the latter methods are too variable. Few manufacturers state in their catalog data the method or methods used in establishing their fan ratings; a small number, however, have followed the practice of publishing ratings of fan capacity based on tests by the more common inaccurate methods and with different ratings for the two sets of conditions, namely, for free air discharge, and for discharge into a short length of duct, usually of a cross-sectional area somewhat greater than that of the fan opening.

If the duct diameter is larger than that of the fan opening, the velocity at the outlet of the duct is necessarily lower than at the fan, and some of the velocity pressure at the fan is converted into static pressure as the air passes through the duct, with resultant gain in capacity.

It is significant that the published ratings for free discharge are invariably greater than those for the somewhat more accurate method of measurement of the discharge into a short length of duct, whereas in all tests which the writer has conducted according to the Standard Code method the results have been the opposite. The principle of conversion of velocity pressure into static pressure by an increase in the cross-sectional area of the stream of air is usually applied to the design of a centrifugal fan, and it seems logical that it can be applied profitably in the installation of a disc or propeller fan. The metal duct on the discharge side of the fan may furthermore decrease air friction by eliminating or reducing eddy currents that exist near the fan at the location where the greatest turbulence takes place, and in some cases the duct may increase the useful capacity of the fan by blocking off the recirculation of air which occurs in certain designs of fans.

The tests herein reported indicate that disc and propeller fans will normally, if not always, have a greater capacity when discharging through a short length of duct of somewhat greater diameter than that of the fan, and that a short duct even of the same diameter as the fan opening will, in the case of an airplane propeller fan, cause an appreciable increase in capacity.

Table 2 shows the capacities of a number of fans for free discharge and for discharge into a short metal duct. The tests from which these results were obtained were conducted according to the A. S. H. V. E. Standard Code method except with reference to the dimensions of the discharge duct. The Standard Code states that "When disc fans are used to blow into a pipe in actual service the performance is different from the performance when mounted in a wall, and consequently allowance should be made for the modified condition. Disc fans for blowing purposes are to be tested with the same standard arrangement of apparatus but in addition there must be placed on the discharge side of the fan a collar of

TABLE 2. FAN CAPACITIES FOR AIRPLANE PROPELLER TYPE AND THIN PRESSED METAL BLADES

Test No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Type of blades	A	A	A	A	A	A	A	A	A	A	A	A	B	B
Number of blades	2	2	2	2	2	3	3	6	6	6	8	8	4	32
Diameter of blade circle, inches	27.15	28.15	28.15	17.85	22.10	26.38	35.62	26.12	35.62	12.05	18.85	20.90	19.00	17.90
Diameter of fan opening, inches	27.40	26.10	26.10	16.50	20.00	26.75	36.50	26.75	36.50	11.00	17.50	19.50	19.50	18.25
Diameter of discharge collar, inches	30	30	26.38	30	30	30	40	30	40	30	30	30	30	30
Length of discharge collar, inches	84	84	40	84	84	84	40	84	40	84	84	84	84	84
Ratio of collar to fan opening diameter	1.10	1.15	1.01	1.82	1.67	1.12	1.10	1.12	1.10	2.73	1.71	1.54	1.54	1.64
Speed for free discharge in revolutions per minute	1745	1694	1710	1964	1700	1765	1750	1750	1730	2044	1704	1150	1154	1190
Tip speed, feet per minute	12400	12500	12600	9190	9830	12200	16330	11970	16200	6450	8400	6300	5740	5580
No-discharge pressure, inches of water	0.255	0.330	0.314	0.200	0.220	0.353	0.725	0.600	1.31	0.236	0.712	0.372	0.472	0.490
Volume of air in cubic feet per minute for free discharge	4770	6230	5970	3025	3970	4225	10970	4930	13580	1530	3380	4150	4450	2590
Volume of air in cubic feet per minute for discharge into collar	6400	7400	6600	3550	5100	5390	14230	6200	17150	1660	4170	4640	5140	2915
Capacity increase due to collar, per cent	34.0	18.5	10.4	17.3	28.4	27.5	29.8	25.6	26.3	8.5	23.3	11.8	15.5	12.5

Note:—*A* denotes blades of airplane propeller type.
B denotes blades of thin pressed metal.

the same diameter as the fan opening and 1 diameter long."

The stipulation of the use of a collar of the same diameter as the fan opening precludes the effect of a reduced velocity of the air in the collar. There is therefore no conversion of velocity pressure into static pressure as in the case where the collar or duct is of larger diameter than the fan opening, and the increase in fan capacity due to the addition of the collar is accordingly lessened.

In the tests recorded in Table 2 it will be noted that the diameter of the collar or discharge duct was the same as the diameter of the fan opening in Test 3 only, and in the other tests ranged up to 2.73 times the diameter of the fan opening. The length varied from 30 to 84 in., a variation which probably has no appreciable effect upon the results.

All values of the electric input to the fan motors and the fan efficiencies are purposely omitted from Table 2 inasmuch as some of the fans which were tested were experimental units in the process of development and the publication of any such data applying to them might be misleading and unfair to the manufacturers. Likewise, the values shown for capacities do not necessarily represent the proper ratings for fans of similar size as they are now being manufactured and are shown here only to aid in later discussion. It is to be noted, however, that in all tests the addition of the collar had little effect upon the power required to drive the fan, in some cases causing a slight increase and in other cases a slight decrease in the power consumption.

Referring to Table 2 it will be noted that in all of the fourteen tests the addition of the duct effected an increase in the capacity of the fan, this increase ranging from 8.5 per cent of the free discharge capacity in test 10 to 34 per cent in test 1. Tests 1 to 12, inclusive, are of fans of the airplane propeller type with cast aluminum blades of various pitch angles, in the majority of designs with the blades of such length as to overlap the fan ring, and in other designs with various amounts of clearance between the tips of the blades and the inside of the fan ring. Test 13 is of a fan of thin blade design such as is commonly used in automobile motor cooling, in unit heaters, and in desk fans. Test 14 is of a disc pressure fan with a large number of sheet metal blades, designed primarily for operation at higher pressures. Fan speeds range from 1150 to 2044 r.p.m., or with tip speeds ranging from 5740 to 16,330 f.p.m.

It is obvious that too many variable factors are involved in these tests to permit of determining accurately from these limited data the most advantageous relationship between the dimensions of the duct and the fan or any definite laws of performance; however, some interesting observations may be made by comparisons of some of the data.

Tests 2 and 3 are of the same fan blades but with motors of slightly different size and with a slight difference in axial clearance between the blades and the inlet face of the fan ring. These differences are probably responsible for the decrease of 260 c.f.m. or 4.2 per cent in the free discharge capacity in test 3 as compared with test 2. In the same tests where the fan was discharging into a duct or collar the collar used in test 2 was of 15 per cent larger diameter than the fan ring, whereas in test 3 the collar and fan ring were substantially of the same diameter. The collar in test 2 permitted of considerable conversion of velocity pressure into static pressure and increased the fan capacity by 18.5 per cent, whereas in test 3 no appreciable conversion of velocity pressure into static pressure was possible and still the collar increased the fan capacity by 10.4 per cent.

As additional evidence of the effect of the collar in test 3 the collar was removed without stopping the fan. This procedure resulted in an increase from 0 to 0.068 in. of water in the pressure in the air chamber from which the test fan was exhausting, thereby indicating that without the aid of the collar the fan under test was unable to balance the capacity of the centrifugal fan which was blowing into the air chamber. The removal of the collar was also accompanied by a noticeable increase in the variation of the needle of the watt-meter which was used to measure the input to the fan motor, thereby suggesting the effect of the collar in dampening out turbulence in the flow of air. Similar evidences were observable in tests of some of the other fans.

In test 10 the collar was 2.73 times the diameter of the fan ring, and the effect upon the capacity of the fan is conspicuously small, namely, an increase of only 8.5 per cent, suggesting that the most advantageous ratio of collar diameter to fan diameter has most likely been exceeded. In general the effect of the collar upon fan capacity, in the case of fans of the airplane propeller type, appears to be more marked in the tests of fans having radial clearance between the tips of the blades and the fan ring than in the tests of fans with blades overlapping the ring. This result is to be expected in view of the fact that in free discharge tests of fans with radial clearance some leakage at the tips of the blades was readily detected, and it is reasonable that a collar on the discharge side of the fan would aid in blocking off this back-flow. No doubt the amount of leakage at the tips of the blades is influenced not only by the clearance but also by the design of the blades, and it is possible that by proper design the leakage may be appreciably reduced.

The jet of air which is discharged by fans of the airplane propeller type is in the form of a cylinder extending a considerable distance from the fan, whereas the discharge from fans with the more common forms of thin sheet metal blades generally flares at a wider angle in the form of a cone. It is to be expected therefore in tests of fans of the latter type that a discharge collar would cause a considerable change in the direction of flow, with consequent increase in fluid friction, so that the collar would show less advantage in the matter of increasing the capacity of the fan. The results of tests 13 and 14 are in line with this expectation in that the increases in capacity due to the collar are only 15.5 and 12.5 per cent, even though the collar was of such a diameter as to permit considerable conversion of velocity pressure into static pressure. These results suggest

the question as to whether a collar of the same diameter as the fan would show any advantage in connection with a fan from which the discharge flares at a wide angle.

Tests 8 and 9 were on six-blade fans of design and dimensions very similar to the corresponding three-blade fans which were tested in tests 6 and 7. A comparison of the results for the six-blade versus the three-blade fans shows that the six-blade fans developed a no-discharge pressure which was in one case 70 and in the other case 81 per cent higher than for the similar sized three-blade fans. If the speeds had been identical these percentages would be slightly increased.

In the matter of capacity the two six-blade fans show free-air capacities 16½ and 24 per cent higher than the free-air capacities of the corresponding three-blade fans. These increases in capacity, however, were gained at the expense of considerable increases in power, in one case an increase of 53 per cent and in the other case an increase of 82 per cent.

The marked increase in power which occurred with increase in capacity is in line with the values shown in Table 1 where it is observed that the power consumed by a fan varies as the cube of the capacity, or as otherwise stated, as the cube of the velocity through the fan. If it is desired to move a certain volume of air it can be accomplished with two lower speed fans with one quarter of the power which would be consumed by one higher speed fan of the same size in performing the same service.

Again referring to Table 1 it will be observed that a 30-in. fan, for example, will deliver under free discharge operation approximately as much air with 0.25 hp. input as will be delivered by an 18-in. fan of similar design with eight times as much, or 2 hp. input. This observation is of prime importance in the selection of a fan for free-discharge service, and it is this feature of relatively large diameters common to the design of airplane propeller fans which accounts in large measure for their low power consumption.

Reliability of Standard Code Method of Testing

In the tests herein reported the measured volume of air entering the air chamber was supplied by a fan discharging through a 24-ft. length of 30-in. duct, and driven by an electric dynamometer. By varying the speed of the dynamometer and in some cases by restricting the inlet of the fan it was possible to control the air supply with sufficient accuracy as to maintain a substantially constant pressure in the air chamber, with variations never more than 0.005 in. of water above or below normal. As an additional check on the air pressure in some of the tests an anemometer was fitted tightly in the wall of the air chamber and it was observed to show no appreciable flow of air either into or out of the chamber, thereby also giving evidence that the measured quantity of air which was being supplied to the chamber was equal to that which was being removed by the propeller fan.

The magnitude of the error which may be introduced in tests by this method due to a failure to maintain zero pressure in the air chamber was investigated by observing the pressure in the chamber when various measured quantities of air, greater or less than the normal capacity of the propeller fan, were being supplied to the air chamber. The results of this test show that a pressure of 0.01 in. of water above or below atmospheric pressure represented an error in fan capacity of 115 c.f.m., which was about 2 per cent of the free discharge capacity of the fan under test. With the ordinary inclined-tube draft gage it is possible to read to a fair degree of accuracy a pressure of one-fifth of this magnitude, which in this instance represented an error in measurement of only 23 c.f.m. or 0.4 per cent of the fan capacity, an error which is much less than that which may be expected in air measurement by the Pitot tube.

One of the most conclusive evidences of the reliability of the Standard Test Code method is shown by observing the pressure in an air-tight chamber when a propeller fan is blowing into the chamber and a similar fan of slightly greater or less capacity is exhausting the air from the chamber; then noting that an equal pressure, but of opposite sign, exists when the fans are interchanged. A test of this sort was occasioned by the statement of Manufacturer *A* that he was confident his fan had a greater capacity than a similar sized fan of Manufacturer *B*, in spite of the results of separate tests which gave Manufacturer *B's* fan a rating of about 1600 c.f.m. above that of Manufacturer *A's*. *A's* fan was set up so as to blow into the air chamber while *B's* fan was exhausting, with the result that a vacuum of 0.067 in. of water was created in the chamber, showing that *A's* fan was not supplying enough air to compensate for that which was being drawn out by *B's*. When the fans were interchanged the pressure in the chamber was found to be 0.065 in. of water, or substantially equal in magnitude to the vacuum which was observed in the former case.

Measurements of Low Velocities By Orifice Meter

In the tests of the various sized fans the same set-up of apparatus was used. This meant that for free discharge or for discharge through an open collar the velocities in the 30-in. measuring duct ranged from 312 to 3500 f.p.m., although most of the values were above 800 f.p.m. When the discharge was restricted in blowing tests the velocity in the measuring duct was even lower than 312 f.p.m., or much too low for satisfactory measurements by a Pitot tube and draft gage. For measurement of the lower velocities it was therefore necessary either to set up a measuring duct of smaller diameter so as to increase the velocities or else to resort to a more sensitive means of measurement.

The drop in pressure through an orifice at the end of the measuring duct was found to serve as a satisfactory means of measurement of the low velocities without in any way interfering with the use of the Pitot tube for measurement of the higher velocities. The orifice was formed by inserting a metal cone in the air-chamber end of the measuring duct. The cone was centered and held rigidly in place in such a position as to restrict the opening an amount suffi-

cient to build up a static pressure of about twenty times the velocity pressure. For any fixed position of the cone the exact ratio of static pressure to velocity pressure was noted when the velocity pressure was sufficiently high as to be determined accurately. It was noted that this ratio remained fairly constant for all velocity pressures of sufficient magnitude to permit accurate readings on the draft gage, varying usually by not more than one per cent from the average value. For velocity pressures which were too low to be read accurately on a draft gage it was therefore assumed that the ratio of static pressure to velocity pressure would remain constant, and the velocity pressure was therefore computed as a definite percentage of the static pressure. Any error in assuming a constant ratio

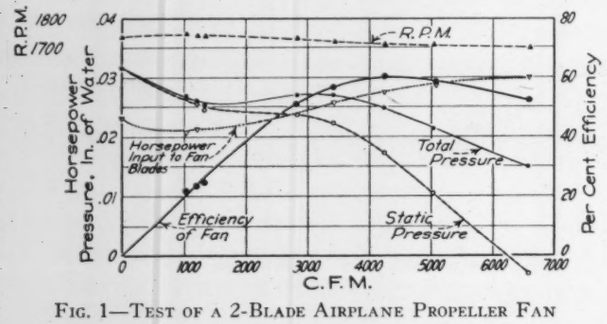


FIG. 1—TEST OF A 2-BLADE AIRPLANE PROPELLER FAN

of static pressure to velocity pressure or, in other words, a constant orifice coefficient for various velocities, is undoubtedly less than the usual tolerance in measurement of air flow.

A cone inserted in the end of the measuring duct is of service not only as an orifice meter but also appears to diffuse air in the air chamber in a satisfactory manner and can therefore conveniently replace the diffuser which is required in the standard arrangement of apparatus.

Airplane Propeller Fans

Within the past few years there have been placed on the market a number of makes of propeller fans in which the blades are of cast aluminum, similar in shape to the blades of an airplane propeller. The manufacturers of these fans contend that in applying the airplane propeller to the propeller fan they have profited by the large amount of study and experimentation which has been involved in the development of efficient types of airplane propellers. They have, however, published very little or no data on the performance of fans of this type aside from their catalog ratings for free delivery, nor have they compared the performance characteristics with those of other types of fans.

It is for the purpose of supplying these data that the results of a test of an airplane propeller fan are shown in Table 3 and Fig. 1. Table 3 shows complete results, a portion of which are shown under test 3 of Table 2. The performance curves of Fig. 1 are plotted from the data tabulated in Table 3, and in shape are fairly representative of the performance of all of the airplane fans which were tested, even though the numerical values of efficiency show better performance than was found in the tests of some of the fans. Slight variations in the shape of the performance curves of some of the other fans have been found to be due chiefly to the influence of the diameter of the collar into which the fan discharged, and in that respect would not represent results strictly in accordance with the standard test code method.

It may be noted that the power input to the motor, in the data shown, is higher than that required by a number of motor-driven fans of similar size. This is due to low efficiency of the motor and not of the fan, inasmuch as the motor which was used in this test proved to have a maximum efficiency of only 59.6 per cent as against efficiencies of at least 70 per cent found in tests of a number of other makes of motors of similar size.

Summary of Results

Table 3 and Fig. 1 show, for free discharge, a capacity of 5970 c.f.m. which is 77.8 per cent of the capacity of a perfect fan, and a corresponding fan efficiency of 47.3 per cent. For discharge into a collar of substantially the same diameter of the fan opening the capacity was found to be 6600 c.f.m. which is 85.4 per cent of the capacity of a perfect fan, and the corresponding fan efficiency was found to be 52.3 per cent.

The maximum fan efficiency of 60.5 per cent occurred when the fan was discharging 4260 c.f.m. against a static pressure of 0.17 in. of water. The power input to the fan blades was a minimum of 0.207 hp. at a capacity of 1050 c.f.m., and a maximum of 0.306 hp. for free discharge. The static and the total pressure curves show a dip in the region of low capacities very similar to that which normally occurs in corresponding performance curves for a multiblade centrifugal fan.

Discussion of Results

The shape of the power input curve is of particular interest in that it shows that when the discharge is closed the power consumption is considerably less than for free discharge. This feature is opposite to that of the majority of disc and propeller fans, which consume the maximum power when the discharge is closed, and would be even more pronounced if the curves of Fig. 1 had been plotted for a constant speed instead of the actual speed of the motor and fan as it occurred in the tests.

No data are here recorded to show the variations in capacity, pressure, and power consumption which occur with variations in speed, but a number of tests at various speeds verify the expectation that, within the limits of accuracy of measurements, the capacity varies directly as the speed, the pressures as the square of the speed, and the power as the cube of the speed. The highest values of efficiency occur in the neighborhood of the maximum capacity, and it is this characteristic which makes fans of this type particularly suitable for moving relatively large volumes of air against small resistances.

In the case of the discharge through an open collar, the static pressure in the collar was found to be 0.030 in. of water below atmospheric pressure,

TABLE 3. TEST OF A 2-BLADE AIRPLANE PROPELLER FAN

ITEM		FREE DISCHARGE	BLOWING TESTS — DIAMETER OF DISCHARGE ORIFICE, INCHES								
			CLOSED	12	14	16	18	20	22	24	OPEN
1	Center velocity pressure, inches of water	0.097	0.00	0.003	0.004	0.005	0.022	0.032	0.049	0.070	0.118
2	Center velocity, feet per minute	1270	0.00	224	259	289	607	733	906	1080	1405
3	Average velocity in measuring duct, feet per minute	1218	0.00	215	249	277	583	704	870	1036	1348
4	Volume of air in cubic feet per minute = average velocity x area of 4.9 square feet	5970	0.00	1052	1217	1358	2850	3440	4260	5070	6600
5	Calculated velocity through fan ring, feet per minute	1605	.....	.....	.....	.....	.....	.....	.....	.....	.....
6	Velocity pressure at fan ring, inches of water	0.154	.....	.....	.....	.....	.....	.....	.....	.....	.....
7	Static pressure in discharge collar, inches of water	.....	0.314	0.264	0.254	0.244	0.236	0.222	0.173	0.105	-0.030
8	Calculated velocity in discharge collar, feet per minute	.....	0.00	278	322	359	755	911	1128	1342	1748
9	Velocity pressure in discharge collar, inches of water	.....	0.00	0.005	0.006	0.008	0.034	0.049	0.076	0.108	0.182
10	Total pressure at fan ring, inches of water	0.154	.....	.....	.....	.....	.....	.....	.....	.....	.....
11	Total pressure in discharge collar, inches of water	.....	0.314	0.269	0.260	0.252	0.270	0.271	0.249	0.213	0.152
12	Horsepower required to move air	0.145	0.00	0.045	0.050	0.054	0.121	0.147	0.167	0.170	0.157
13	Electric input to motor, watts	385	290	263	275	278	300	325	347	365	380
14	Equivalent horsepower input to motor	0.516	0.389	0.353	0.369	0.372	0.402	0.435	0.465	0.489	0.510
15	Speed of fan and motor in revolutions per minute	1710	1732	1743	1738	1737	1728	1718	1710	1707	1700
16	Efficiency of motor, per cent	59.3	59.1	58.6	58.9	58.9	59.2	59.4	59.5	59.6	59.3
17	Horsepower input to fan blades	0.306	0.230	0.207	0.213	0.219	0.238	0.258	0.276	0.291	0.302
18	Overall efficiency of fan and motor, per cent	28.0	0.00	12.6	13.5	14.5	30.1	33.8	35.9	34.7	30.9
19	Efficiency of fan, per cent	47.3	0.00	21.5	23.4	24.6	50.9	56.8	60.5	58.4	52.3
20	Volume of air in cubic feet per minute for 100 per cent fan efficiency	7670	.....	.....	.....	.....	.....	.....	.....	.....	7725
21	Actual capacity in per cent of that of a perfect fan	77.8	.....	.....	.....	.....	.....	.....	.....	.....	85.4

Diameter of blade circle, 28.15 in.
Diameter of fan opening, 26.10 in.
Discharge collar, 26.38 in. diam. x 40.0 in.

Corrected barometer, 29.03 in. mercury at 75 F.
Velocity coefficient for center of pipe, 0.96.
Velocity, feet per minute, $4092\sqrt{\text{V. P.}}$

thereby indicating a contraction of the stream of air within the collar. If this negative pressure had been neglected, on the assumption that zero static pressure existed at the outlet of the collar, the result of fan efficiency would be increased from 52.3 to 62.7 per cent, but this value would not be consistent with the other values of efficiency which were based upon measured pressures inside of the collar.

The efficiency of 47.3 per cent for free discharge is considerably higher than the efficiency claimed for any centrifugal fan for operation under similar service, such values for centrifugal fans usually ranging from 20 to 35 per cent. Similar tests, however, on some of the designs of airplane propeller fans showed efficiencies even lower than the values quoted for centrifugal fans, the variation depending largely upon the amount of leakage at the tip of the blades.

The maximum efficiency which occurred when the fan was discharging against resistance, namely, 60.5 per cent, is not quite as high as the maximum efficiency found in tests of many centrifugal fans, even though it represents much better performance than that which was found in tests of some of the other propeller fans.

In comparing the performance of propeller and of centrifugal fans it should be observed that in the selection of either type for any certain installation a wide range in power consumption is possible, depending upon the velocity at the outlet of the fan, and in order to maintain as low a velocity at the outlet of a centrifugal fan as in the case of a propeller fan, the area of the outlet must be of the same size as the entire opening of the propeller fan, and the entire housing of the centrifugal fan therefore becomes relatively large.

Conclusions

The claim is often made that the airplane propeller is the most efficient device yet known for the moving of air. If this statement is limited to the free discharge of air or to operation at or near the maximum capacity of the airplane propeller fan it is apparently true. However, the efficiency of a propeller fan is dependent not only upon the design of the propeller itself but to a great extent upon the means for preventing leakage at the tips of the blades.

A collar or short duct on the discharge side of the fan increases the capacity and efficiency, not only by reducing leakage but also by reducing eddy currents, and if the diameter of the collar is somewhat larger than that of the fan the performance is still further improved by the resulting reduction in velocity pressure.

Propeller fans for operation against the higher static pressures have merit with reference to small space requirements and relatively low cost as compared with centrifugal fans but do not show as high efficiencies as are found in tests of centrifugal fans.

Analysis of the fan performance as published by the manufacturers of a considerable number of disc and propeller fans shows unreasonable values of fan capacity and efficiency. The calculated value of fan efficiency and the comparison of the capacity rating with the computed capacity of a perfect fan are serviceable checks on the reliability of catalog data.

The standard method of testing disc and propeller fans as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers* is a reliable method of testing, and shows important features of fan performance which are likely to be overlooked in tests by the older and less accurate methods.

Comments on Design and Rating of Direct Radiation

By Konrad Meier, Winterthur, Switzerland
MEMBER

CONDITIONS in the field of direct heating bring out sharply the diverging tendencies of designer and engineer. In evolving new designs and devices the maker naturally considers production and listens to popular demand, while the engineer, who does the planning and knows the legitimate needs, will set another standard, based on efficient and perfect service to the client. In view of developments, particularly in regard to rating, which is significant of the situation, it should be useful to restate the case as seen from the engineering side.

The A. S. H. V. E. GUIDE 1928 contains a series of tables for heat emitted by direct radiation, which covers the principal patterns of the column and wall types. The rated square feet, probably approximating the actual, appear together with the capacities in B.tu. as tested by a recognized authority. Unfortunately, this compilation is not extended to the tubular and other designs, because generally accepted engineering data are not available. A table for one of the tubular patterns was published early in 1928 by A. H. Schroth,[1] wherein capacities are given alongside of the ratings in equivalents of 240 B.tu per square foot. This at least permits the calculation of the heat emitted per square foot of surface for comparison with the well established figures for the coefficient U. However, most of the recent literature on that now current style and of fin tube metal heaters gives only the so-called square feet based on the conventional 240 B.tu, the manufacturers thus leaving the buyer in doubt as to the actual surface. But after all, it is the latter, as the absolute measure of quantity, which should also be given, with the performance as additional information for the engineer to be used with such conversion factors and judgment which the case may demand. Indeed, there is no valid reason why this could not be done for heating as for other apparatus. Pumps are not sold by the gallon, but according to sizes, for which the capacities are also given under stated conditions.

In Mr. Schroth's table, the higher rate of emission for short units is duly taken into account. This point is ignored by other manufacturers, evidently because they do not like the idea of different ratings for identical sections. It would certainly reveal the fallacy of the method, by which the variable heat emission is expressed in square feet, instead of simply giving the actual surface and its capacity in the recognized measure for heat and for standard conditions of testing. The latter way plainly gives the variations inherent to the kind of surface and will naturally lead to the consideration of the other factors bearing on the case.

Figs. 1 to 5 illustrate the variability of the so-called square foot. The curve for tubular radiation (Fig. 1) is based on the table referred to and issued by a manufacturer, while all the others are derived from the data of the late Prof. John R. Allen and Prof. F. B. Rowley, adopted by the Research Laboratory and published in the A. S. H. V. E. GUIDE. It will be noted that the curve for the tubular pattern is similar to those for the column radiation, Fig. 2. For both, the end sections show greatly increased heat emission, the excesses running from 46 to 71 per cent over that of the intermediate or of units of about 20 sections and over. The variations within the number of sections frequently used average from 5 to 6 per cent for low and 12 to 15 per cent for high patterns. The factor of length is therefore not altogether negligible, particularly for the greater depths. Incidentally, the curves recall the variations due to height alone, which run from 8 to 18 per cent. The variations between one and four column patterns are 23 per cent for 38 in. and only 12 per cent for 20 in. height. For wall radiation the differences owing to height do not exceed 7 per cent and those for length are negligible, while the average is about 20 per cent higher than the old standard two-column. These unquestioned figures, ranging roughly from 200 to 350 B.tu, refer to actual surface. Hence the divergence of the square feet as now sold on the 240 B.tu plan must cover an equal range for plain cast iron radiation alone. This means very material deviations from the true surface measure. Whether these are likely to be stated fairly in trade literature under stress of competition is for the reader to judge. For standardized articles the buyer will soon enough find out the facts, but he is virtually expected to take on good faith the ratings for the numerous odd sizes of tubular radiation and for the concealed surfaces of still greater individuality. Evidently, the tendency is to drift away from standardization, which makes it more and more troublesome to obtain authentic data on performance. Simple comparison is becoming practically impossible and the conscientious engineer is often at a loss to pass judgment on equipment offered to his client and to obtain a proper basis for his calculations and allowances.

A further source of uncertainties, which has increased with the advent of recent heating methods, lies in the manner and the details of installation. This applies, for instance, to the semi-exposed sur-

[1] *Heating and Ventilating Magazine*, February, 1928.

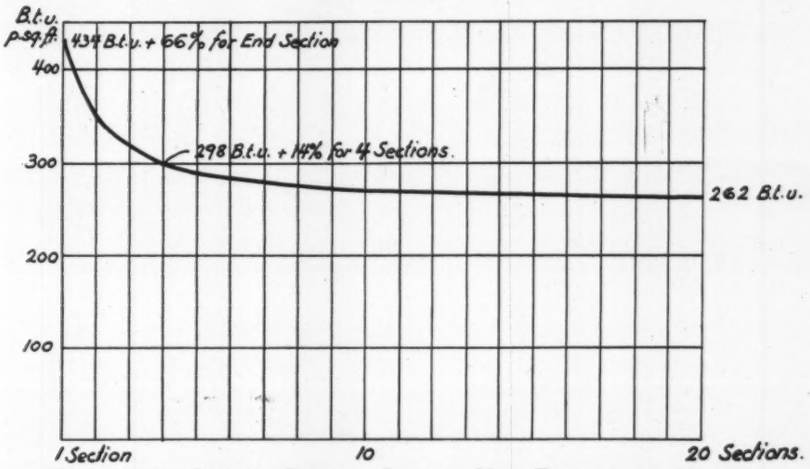


FIG. 1—CURVE SHOWING RELATION BETWEEN HEAT EMISSION AND NUMBER OF SECTIONS FOR A 5 TUBE, 38 IN. RADIATOR

(Based on table from Heating and Ventilating Magazine, Feb., 1928).

faces, for which the ratings appear to be the same, whether grille-work is used on top only or top and bottom and with the sides inclosed or not. There must be differences, depending moreover, on the depth and height of the pattern selected.

In a still greater measure this is true of the screened radiation. The numerous tests on the effect of inclosures have shown that the heat emission is materially affected by the arrangement and the proportions of the openings, the spacings and other features more or less beyond control of the engineer. It is also well known that the allowances should vary according to style and height of the radiator, although this point is generally ignored owing to lack of data. Indeed, it seems impossible to set up any reliable rule for coverings, except that of using judgment and of providing an ample margin of safety necessarily at the expense of the owner.

The situation is similar in regard to the invisible or concealed radiation. In reality, these devices are distinctly air heaters for which the capacity is largely governed by flue action. But the latter must vary to a considerable extent according to local conditions not always known beforehand by the manufacturer or the engineer. Moreover, the closely spaced and inaccessible fin surfaces are liable to gather dust, which in time reduces the air volume and efficiency. Finally, the absorption and dissipation of heat by the adjoining walls is at least a retarding factor and a hindrance to good service, quite aside from the losses due to hot strata overhead and the consequent reduction of room heating efficiency, which has not yet been fully investigated for these devices. In view of these facts, manufacturers' data should not be used indiscriminately.

It has been proposed that radiation be rated according to room heating effect. This looks plausible in theory, but would involve the inclusion of factors outside of design and beyond the control of the manufacturer. Although inherent to some extent in the form and proportions of the surface, it is largely a matter of distribution, location and operation. This quite aside from the vagaries of building construction, which make it uncertain enough even for the contractor and engineer to assume any responsibilities for results. In any event, room heating efficiency could never be a logical basis for rating in the accepted sense, since it is not a measure of capacity for stated conditions, but rather one for variable conditions and therefore likely to become a source of confusion. It should be expressed by itself, as a ratio of the heat utilized to the heat actually expended,

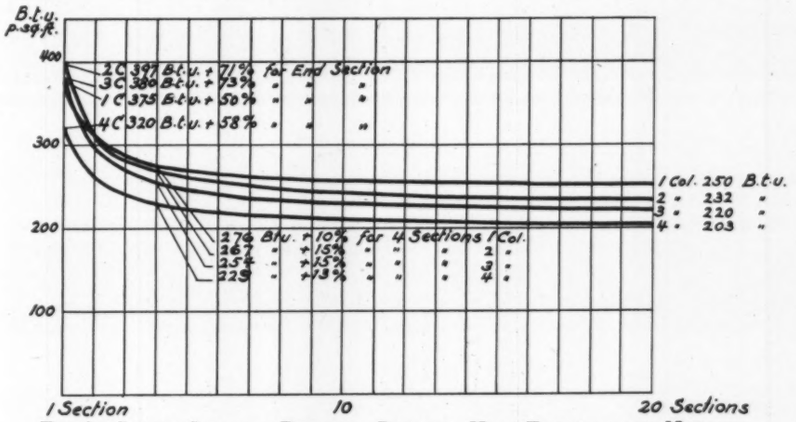


FIG. 2—CURVES SHOWING RELATION BETWEEN HEAT EMISSION AND NUMBER OF SECTIONS FOR 38 IN. COLUMN RADIATORS

(Based on A. S. H. V. E. Research Laboratory Data, Chapter 2, A. S. H. V. E. GUIDE, 1929.)

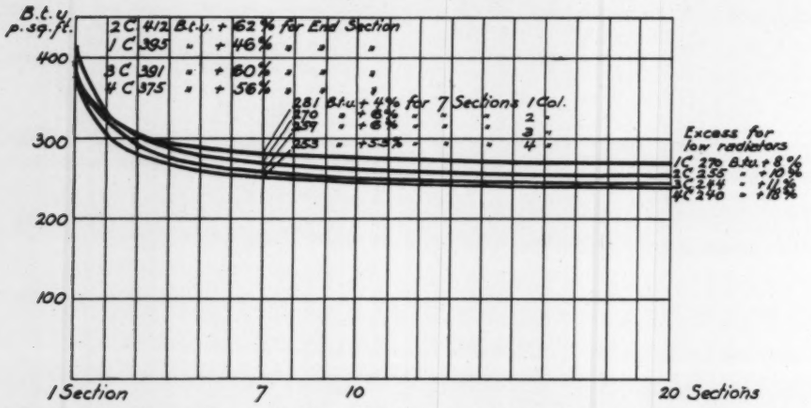

B.t.u. p.sq.ft.
400
300
200
100
2C 412 B.t.u. + 62% for End Section
1C 395 " + 46% " " "
3C 391 " + 80% " " "
4C 375 " + 56% " " "
281 B.t.u. + 4% for 7 Sections 1 Col.
270 " + 8% " " " 2 "
257 " + 6% " " " 3 "
253 " + 3.3% " " " 4 "
Excess for low radiators
1C 270 B.t.u. + 8%
2C 255 " + 10%
3C 244 " + 14%
4C 240 " + 18%
1 Section
7
10
20 Sections

FIG. 3—CURVES SHOWING RELATION BETWEEN HEAT EMISSION AND NUMBER OF SECTIONS FOR 20 IN. COLUMN RADIATORS

(Based on A. S. H. V. E. Research Laboratory Data, Chapter 2, A. S. H. V. E. GUIDE, 1929.)

resulting from the selection and disposition of the surface, which does demand consideration. Direct heating, like hot-blast and other systems, can be applied to more or less advantage. To provide for the best possible service to the owner is one of the principal duties of the engineer, who needs more authentic information for that purpose.

The real causes of the present unsatisfactory situation in the radiator field can be traced at least partly to the old superficial rule of judging bids by square feet alone, irrespective of their value. It is this practice which has led contractors to offer a maximum of surface at minimum cost without much regard to fitness for the case and accounts also for certain discrepancies between the actual and the rated catalog surfaces that have long since bothered the engineer. The new method of rating is an attempt to legitimize these discrepancies by putting them on a scientific basis, thereby retaining the customary measure, but giving it variable valuation. Unfortunately, with recent developments this way of rating has become more and more difficult and uncertain. Judging by experience and present indications there is not likely to be a correction, but rather an increase of the former discrepancies. Under such conditions, of course, this method is anything but conducive to efficiency and economy in application. Surely, the clear-cut statement of the actual surface together with its performance and other essential points would seem to be more than ever welcome, at least to the profession. In reality, it should be not less acceptable to the other side, since eventually it should lead to the production of true square feet of demonstrable capacity and desired qualities.

Good engineering involves the elimination of uncertainties. A main source of these lies in the difficulties of the variable rating. However, this is not at all a condition that need be accepted in the opinion of the writer. A study of the data at hand will show that radiation can be designed having practically the same heat emission for all sizes. While it would not pay to evolve a new type on that score alone, this might be justified if other desirable qualities can be assured in combination.

To reduce the variations in length, the greater part of the surface must naturally be in front and back, rather than on the sides or flanks facing each other and sideways. This seems but reasonable. The surface on the whole will thereby become more effective and proportions may be secured that will suit the usual spaces under windows, occupying their length and protruding but little into the room. The variations due to height are likewise amenable to reduction if the air circulation is adequately favored.

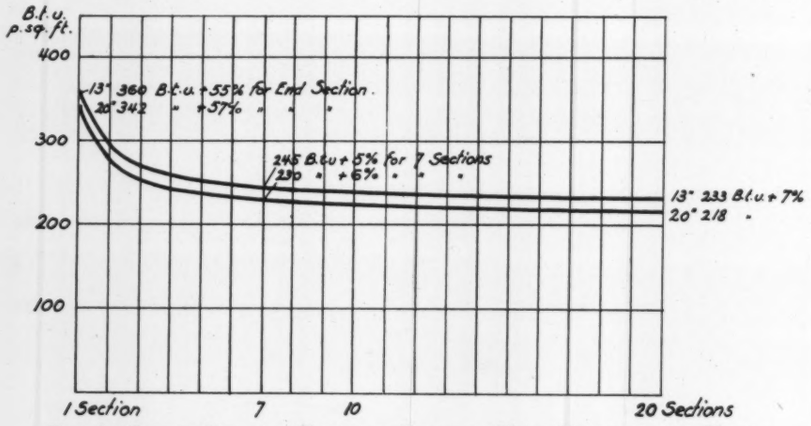


FIG. 4—CURVES SHOWING RELATION BETWEEN HEAT EMISSION AND NUMBER OF SECTIONS FOR 13 IN. AND 20 IN. WINDOW RADIATORS

(Based on A. S. H. V. E. Research Laboratory Data, Chapter 2, A. S. H. V. E. GUIDE, 1929.)

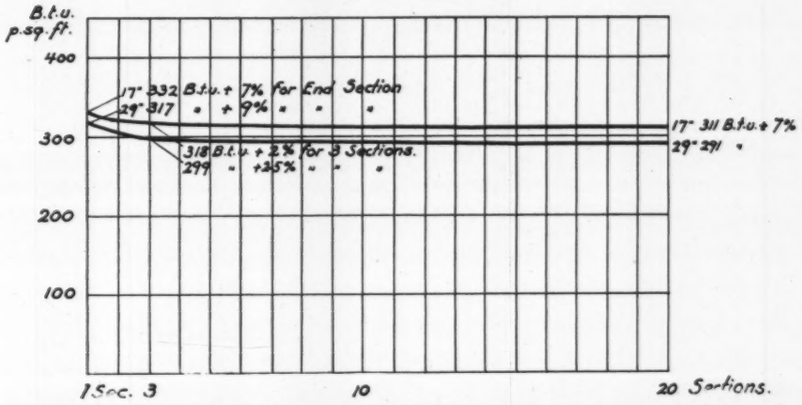


FIG. 5—CURVES SHOWING RELATION BETWEEN HEAT EMISSION AND NUMBER OF SECTIONS FOR 17 IN. AND 29 IN. WALL RADIATORS
(Based on A. S. H. V. E. Research Laboratory Data, Chapter 2, A. S. H. V. E. GUIDE, 1929.)

Increased volumes of lower temperature near the top will keep up the emission for the taller units and at the same time give more agreeable effect. This may be secured by open and unobstructed vertical air channels. When applied to the side facing the wall, where only the heat of convection is utilized, they will carry the heat freely and without undue losses to the masonry to where it will meet indraught and downdraught from the windows. This will also prevent warmer strata overhead, especially under wind pressure, the heat then properly reaching the occupied zone. The radiant heat, as will be noted from the curves of single column and wall radiation, also has a steadying influence on the rate of emission. Hence the side of the radiator facing the room should be designed for radiant effects, which helps to warm the floor, particularly near the windows where space is valuable and the direct action of this radiant heat will also greatly add to comfort if surface temperatures are reduced as they should be for hygienic reasons and as maintained naturally in first class water heating.

Indeed, one of the important, but little heeded requirements of wholesome heating, is the elimination of all unpleasant effects, particularly the irritation caused by dried dust, so often taken for excessive dryness of the air. Clean surfaces of moderate temperature seem to be the only safe and sensible way of assuring this quality with direct radiation. The additional surface required to that end is a useful improvement and a good investment, while the screening often demanding an equal amount and involving other costs, is always a decided detriment.

To design heating surface for high and constant heat emission therefore leads to progress in more than one direction. It results in proportions suited to spaces beneath windows for which neither column nor tubular and wall patterns are specially adapted, thus filling a distinct gap in the patterns available. Being essentially a wall radiator, attachment to the wall seems natural as well as desirable for unobstructed flooring. The supports should form part of the radiator in order to assure the right distance from the wall for free circulation and for easy cleaning. Aesthetic quality requires further that each radiator appear as an individual unit. Taking advantage of the distinctive treatment of the front, this is readily accomplished by modified end sections, which will also have tappings for the supports and for practically invisible connections in the rear from run outs in flooring or behind the bases. This secures the neatest installation which should in the great majority of cases obviate the old objection to exposed radiation, not in the least due to unsightly accessories. But then, finished installations are not possible without due attention to detail, which must begin with design.

With service as the governing consideration, it is therefore not only feasible to attain higher efficiency by turning the typical section endways, but to provide B.tu at moderate and possibly at even lower cost. The rating, moreover, will be fairly equal for all sizes and it will be dependable. As a finished product inducing neatest installation and dignified treatment, it gives no excuse for screening, which is not a solution but an evasion of heating problems. Finally, in so far as the location of radiators beneath windows is favored, the best room heating effect is assured. Thus all of the uncertain factors in rating and application are practically eliminated by plain exposed radiation, when designed for highest thermal, hygienic and aesthetic qualities. And these qualities are attainable at no unreasonable cost. In fact, the consistent use of heating surface beneath windows would permit a reduction of patterns, if these were specially suited. Two or three heights in one depth will suffice in all ordinary situations and serve also, if need be, on inside walls. For these few sizes at least, standardization in general dimensions might easily be brought about. It would not involve the individuality of design. Competition in regard to the latter might rather be stimulated and prove to be valuable in evolving attractive radiation, thus helping the cause in general. Hygienic quality is coming to the fore. It has rightly led to the control of temperature and purity of the air in blast heating. Such control is not practicable with concealed radiators, but the desired results are attainable with clean exposed surfaces having mild radiant effect and producing air currents of moderate temperature. The

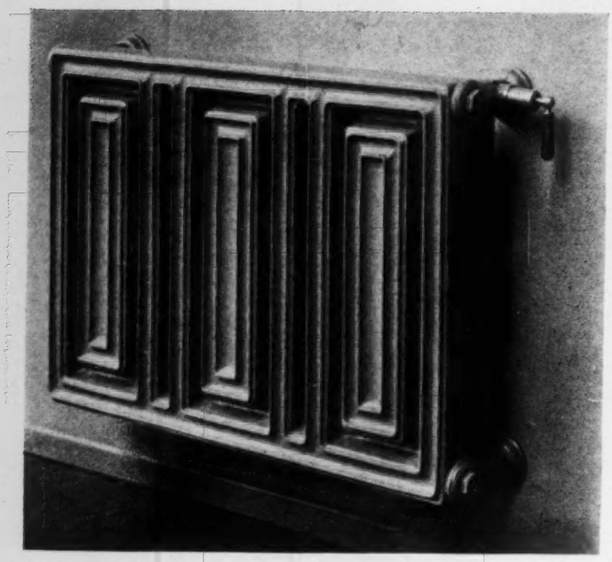


FIG. 6—COLUMBUS RADIATOR MADE IN SWITZERLAND SHOWING CONNECTIONS AND SUPPORTS

choice of type of system should lie between these.

Steam heat is no longer installed in first class buildings in Central Europe, principally for hygienic reasons. Also, exposed radiation is gaining ground, not merely because of financial necessities. It is the trend for simplicity and truth, doing away with all sorts of frills and favoring the saner practice. This puts the heating man under the obligation of turning out truly neat and finished work. But to bring the standard of installations nearer to what is done in other lines, would certainly do no harm.

Radiation on the plan outlined was developed several years ago with the idea of combining such high qualities as would not exclude mass production, but should rather stimulate it and eventually result in lower cost. Figs. 6 and 7 represent two patterns, introduced in Switzerland. One of the designs is a slab section which form seems appropriate for radiant effect. The rate of heat emission is fully equal to that of typical single column patterns, while the space occupied is considerably smaller, also compared with the tubular style. Weight and water contents are about equal to the latter. Only three heights are needed of each design.

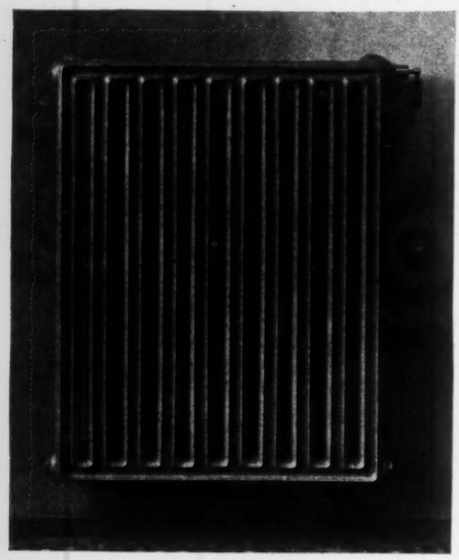


FIG. 7—COLUMBUS RADIATOR MADE IN SWITZERLAND SHOWING REGULATING VALVE AND CONNECTIONS INSIDE OF WALL

In spite of the present diverging tendencies in the United States, true progress would seem to lie in a similar direction, that is, in keeping up the development of clean and neat cast iron radiation, which needs no excuse or concealment, if judiciously selected to fit the space available. Design for service will also lead to standardization and the question of rating will then practically solve itself. This may be said to represent the legitimate demand of the engineers in the interest of their clients. Actual experience has shown that it involves no impossible task for the manufacturer and that such a product would be welcome to progressive architects, engineers and owners. The wide field of application certainly assures the economic production for the benefit of all.

Warming of Walls

The following interesting analysis of the heat capacity of walls entitled "The Warming of Walls" by A. F. Dufton, *M. A., D. I. C.,* of *The Building Research Station* appeared in the November 1927 issue of The *London Edinburgh and Dublin Philosophical Magazine:*

"1. It is well known that some rooms are not readily warmed. Complete analysis of the warming of a room is not easy. The influence, however, of the fabric of a wall is shown by the time taken to warm the surface when heat is supplied to it at a constant rate.

"2. For simplicity a homogeneous wall (of thickness d, conductivity k and diffusivity h) initially at uniform temperature is considered. The rate of heating chosen is $2\ T\ k/d$, twice that necessary to maintain the desired temperature difference T between the faces of the wall, and the temperature of the second face is assumed to be constant.

In time t the temperature of the wall at distance x from the heated surface increases by

$$T\left\{ 2 - \frac{2x}{d} - \frac{16}{\pi^2} \sum_{0}^{\infty} (2n+1)^{-2} e^{-(2n+1)^2 \pi^2 ht/4d^2} \cos (2n+1)\pi x/2d \right\}.$$

The rise at the surface, the limit of this as x tends to zero, is

$$T\left\{ 2 - \frac{16}{\pi^2} \sum_{0}^{\infty} (2n+1)^{-2} e^{-(2n+1)^2 \pi^2 ht/4d^2} \right\}$$

and attains the value T when

$$\frac{16}{\pi^2} \sum_{0}^{\infty} (2n+1)^{-2} e^{-(2n+1)^2 \pi^2 ht/4d^2} = 1,$$

approximately after a time $d^2/5h$."

The World Engineering Congress

BECAUSE of the wide scope of the subjects covered and the vital influence brought to bear on the present aspects of international engineering, the World Engineering Congress held in Japan October 30 to November 7, 1929, probably outranks all other engineering conferences. The World Engineering Congress was held under the auspices of the Kogakkai, which is the engineering society of Japan, and is a federation of twelve technical societies of that country.

Many engineering congresses have been held in the past in Europe and the United States, but this is the first occasion on which the Far East has convened such a gathering. Japanese engineers had long hoped for and awaited the opportunity for such an international congress to be organized under their auspices. Both the Imperial Japanese government and the general public, including important societies and eminent industrial concerns, readily gave their support, the government appropriating a special subsidy and the public making a general contribution toward the necessary funds. The Preparation Committee was under the chairmanship of Baron K. Furuichi, President of Kogakkai.

The A. S. H. V. E. official delegates appointed by the Council, were W. H. Carrier, Newark, N. J.; Shozo Saito, Tokyo, and Kunisuke Sekido of Nakano. Mr. Carrier, who is second vice-president of the Society, and president of the Carrier Enginering Corp., presented a paper entitled *Control of Humidity and Temperature as Applied to Manufacturing Processes and Human Comfort,* which was published in the November, 1929 issue of *Heating, Piping and Air Conditioning.*

A most cordial welcome was extended to the delegates and guests of the Congress in the opening address by the Prime Minister of Japan, His Excellency Hamaguchi, who spoke in part as follows:

We in the Orient have from ancient times a saying that nothing is more pleasant than to have friends come from a distance by no means does justice to the case of the visiting delegates; some of you have come across the seas from lands tens of thousands of miles away, nor is the word "pleasant" adequate to express what we feel. It is perhaps a little more the truth to say that we are deeply stirred by irrepressible sentiments of joy and appreciation.

A reception was given in the honor of the Overseas Delegates on November 3, 1929, by the *Japanese Society of Domestic and Sanitary Engineering,* the principal speakers being S. Kitaura, president of the organization, B. Mano, former president, and Shozo Saito.

Dr. Kitaura spoke as follows:

Allow me to say a few words in order to express our hearty welcome and thanks to all our guests who have come to this banquet held in Japanese style for the entertainment of all those Overseas delegates and members participating in the World Engineering Congress, who belong to the societies in their own countries intimately related to the *Japanese Society of Domestic and Sanitary Engineering,* of which I have the honor of being the president.

Most of your papers related to our Society are not yet read, up till this day, at the meetings of the Congress, but I have already been enabled to look into their abstracts or advanced prints, and found them all to be very valuable reports, and when published they will no doubt benefit and keenly interest the international scientific circles. I beg therefore, by this opportunity, to thank you most sincerely and ask you to kindly convey our profound feelings of gratitude to the members of your own societies at home.

I ventured to invite you this evening to a dinner purely Japanese, and I hope this may not be wholly distasteful to you, though so different from the customs and manners to which you are accustomed at home, and that this may be an interesting experience on your journey.

Gentlemen, please make yourselves as comfortable as possible here.

Dr. Mano spoke in part as follows:

The President, Mr. Kitaura, gave me a privilege to say a few words on this occasion, which I consider a great honor.

First of all, I have to thank you, our esteemed guests, for sparing your most valuable time for us this evening.

What I want to say is this, that our President and officials of this Society have taken great pains in entertaining you; you have been to several dinners, to many garden parties, evening parties as well as other gatherings and we wanted to show you something new in this line and after consulting many programs of this kind, we came to the conclusion that it would be best and wise for us to give you a Japanese dinner in quite a Japanese fashion.

Our dishes may not meet with your taste; our singing girls may not be able to speak English, but we think you will find the singing girls entirely charming and as our Japanese dinner is concerned, you will find it quite new. * * * Our Society, of which I am one of the founders, was born only a few years ago and we look upon you as fathers and grandfathers as far as the existence is concerned, and as specialists of domestic and sanitary engineering you rank as teachers and professors, and therefore this meeting is the gathering of our family, being more than that of friends.

We hope therefore that you will take this as a family party and please put your minds at ease (unfortunately I can't ask you to sit at ease as you have difficulty in sitting in our Japanese fashion).

We prepared this gathering for you this evening, hoping you will fully enjoy our social way of extending hospitality to you all. By your so doing, we shall feel more than satisfied.

Mr. Saito spoke as follows:

Allow me to introduce myself to you. My name is Shozo Saito. I have been for years a member of the *American Society of M. E.,* H. V. E., *R. E.* and the *Institute of H. V. E.* of London and also a member of the *Society of Domestic and Sanitary Engineering.* But, for tonight, I only wish to speak to you personally and not officially.

It is really a greatest honor and pleasure for me to enjoy this evening with you who have come out so far away, and to my unfathomable delight the World Engineering Congress took place here in Japan for this time and I was able to have the unique opportunity to be acquainted with so many prominent personages of world-wide known. And we are all here together to spend this wonderful evening to augment our friendship.

Indeed, I am very much grateful for your presence this evening at this Reception Party. And this party is to be greatly ascribed to the patronage and support of those companies, the names of which are given in the leaflets distributed among you.

At an *After Meeting of the Reception Party* held on December 11, 1929, good wishes were extended to the Overseas guests for a Merry Christmas and a Happy New Year. A card bearing the signatures of the individuals attending this meeting, is reproduced here-

with. Among the signatures are those of S. Kitaura, President of the *Society of Domestic and Sanitary Engineering,* and Dr. Ritaro Hirota of the Reception Committee.

Many interesting papers relating to heating and ventilation were delivered at the Congress. Of the 544 papers submitted, 319 came from Japanese sources, and 225 from foreigners. The following are verbatim ab-

stracts by Mr. Saito of some of the papers relating to heating and ventilation:

The Development of the Heating System in Japan, by Kunisuke Sekido.

In the preparation of this brief note the writer tries to give to the readers an insight into the different stages of development which Japan has gone through in her heating system. In undertaking this he makes use of the close and intimate connections he has had with this problem in Japan for over thirty years. He depicts various phases of this branch of engineering practice as has obtained in Japan since its first introduction in about 1870, in five sections as follows: Section I, Introduction; Section II, Steam and Hot Water Heating; Section III, Warm Air and Hot Blast Heating; Section IV, Cooling Systems; and Section V, *Ondor* (Korean Native Method of Heating).

Ventilation of a Railway Tunnel, by Director of Improvement Section of the Department Railway.

The combustion gases produced by steam locomotives in railway tunnels not only corrode the track materials and disintegrate the concrete lining, but also threaten the health of engine-men and tunnel laborers, consequently lowering their efficiency. This is especially severe in our Government Railways as we have many long and steep grade tunnels under heavy traffic, the precise data of which are given in the papers attached. Hence the ventilation of a long railway tunnel is one of the most serious and difficult problems presented to the railways.

Last year we provided the first ventilation plant at the Higashiyama tunnel on the Tokaido line and this year, the second one at the Kobotoke tunnel on the Chuo line.

This report describes mainly the ventilation plant at the Higashiyama tunnel in the following order:

1. Determination of the reasonable quantities of various gases produced by coal combustion, and quotation of the numerical examples obtained by the actual observation made at the existing tunnels Kobotoke and Utuo on the Chuo line.
2. Description of the properties of gases in the tunnel and their influence upon human body as well as upon track materials and tunnel structures.
3. Description of various methods of tunnel ventilation adopted by other railways in the world.
4. Description of the ventilation plant at the Higashiyama tunnel by the following items:

(1) The locality and meteorological condition of the tunnel.
(2) Details of the ventilation plant installed.
(3) Quantities of poisonous gases produced in the tunnel.
(4) Determination of the ventilation velocity, wherein the several methods of determining this velocity are described, and further, the determination of the ventilating velocity, i. e., five meters per second in consideration of the health of tunnel laborers and the local conditions of the Higashiyama tunnel as well as the operating time of the fan, is accounted for.
(5) Determination of the necessary horsepower. As mentioned above, ventilating velocity is determined, and then the necessary horsepower of the fan is calculated by the following formula:

$$h = \text{resistance} = \frac{c\ u\ l\ v^2}{2\ g\ A}$$

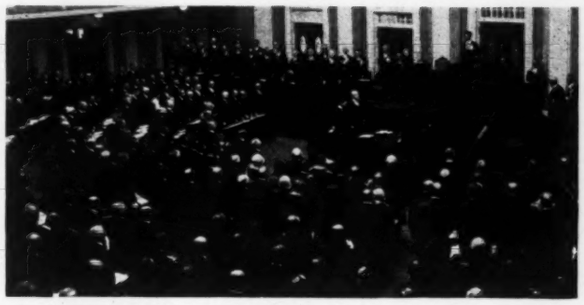
$$\text{Necessary hp.} = \frac{\text{quantities of air per second} \times \text{resistance}}{75}$$

M. hp. = 1.6 × N. hp. and 150 hp. adopted

(6) Method of the operation of the fan.
(7) Actual effect of the ventilation plant upon tunnel laborers, track materials and tunnel structure.
(8) Construction cost, operation cost and amount of money reduced by ventilation plant.
(9) Ventilation test at the Higashiyama tunnel. The coefficient of the ventilating friction is very uncertain and it depends upon various factors. But to our great regret we have no reliable value of this coefficient for the railway tunnel. And we executed actual observation at the Higashiyama tunnel to determine the coefficient of friction, on completion of the ventilation plant at this tunnel. By this observation the coefficient of the friction in the Higashiyama tunnel has been decided to be 0.00674.

Ventilation of Vehicular Tunnels by Ole Singstad.

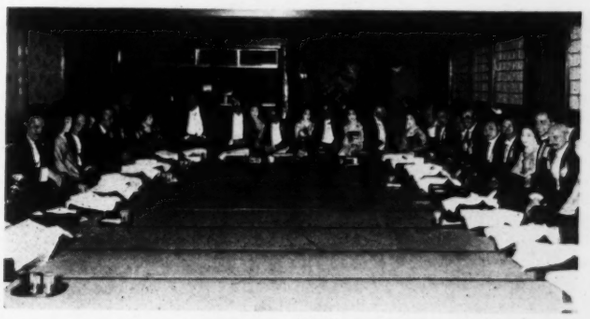
Tunnels for vehicular traffic are of comparatively recent date. Before the general advent of the automobile, such tunnels depended upon natural draft for ventilation. Those of more recent date have been provided with mechanical ventilation. The outstanding example of the latter class is the Holland Tunnel under the Hudson River at New York, consisting of two tubes, each equipped with a fresh air duct under the roadway and a vitiated air duct above the ceiling. These ducts are connected to the roadway area at frequent intervals by lateral flues or ports, and lead to a system of blower and exhaust fans housed in four large ventilation buildings. This system provides an air move-



H. I. H. Prince Chichibu Addressing Meeting at Opening Ceremony in the Parliamental House, Tokyo, Oct. 29, 1929

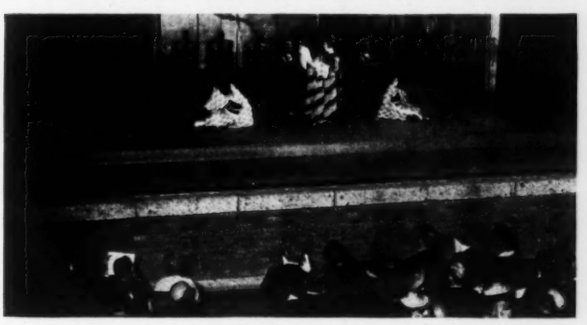
Address of H. I. H. Prince Chichibu at the Opening Ceremony in the Municipal Auditorium, Tokyo, Oct. 29, 1929

H. I. H. Prince and Princess Chichibu's Garden Party at Shinjuku Imperial Garden, Tokyo, Oct. 31, 1929



Meeting of International Executive Council at Koyo-kan (Maple Club), Tokyo, Oct. 29, 1929

American Night at Imperial Hotel, Tokyo, Oct. 31, 1929



Japanese Dance Performance, Baron Okura's at Home, Imperial Hotel, Tokyo, Oct. 31, 1929

ment across the section of the tunnel instead of a longitudinal one as is the case in all tunnels constructed previously. The Holland Tunnel system of ventilation thus obviates the necessity of high velocity air currents in the tunnel roadway.

In planning the ventilation system of that tunnel it was found that very few authoritative data were available regarding the amount and composition of exhaust gases from automobiles, the physiological effects of such gases on human beings and power losses in concrete ducts, flues, elbows and air ports. Previous experiments on exhaust gases were not conclusive and the reliability of the existing formulae on power losses was questionable.

It was therefore considered necessary to conduct independent experiments to establish such fundamental data before undertaking the design of the ventilation system of The Holland Tunnel.

The research work was carried out under a contract between the New York State Bridge and Tunnel Commission and the New Jersey Interstate Bridge and Tunnel Commission, jointly in charge of constructing the tunnel, and the United States Bureau of Mines, whereby the Bureau was to carry on the investigations in accordance with the program laid down by the Engineer of the Tunnel Commissions.

Investigations to determine the amount and composition of exhaust gases from motor vehicles were conducted on more than one hundred vehicles, under many conditions of service and road.

Experiments to determine the physiological effects of carbon monoxide were conducted in concentrations up to 10 parts of carbon monoxide in 10,000 parts of air. It was established that 4 parts of carbon monoxide in 10,000 parts of air was the maximum concentration permissible in the tunnel atmosphere, and that exposure to such a concentration should not exceed one hour.

The research work to determine the friction losses in ducts, bends, etc., and the power required to furnish the requisite ventilation, were conducted on models of the tunnel ducts, flues and ports up to half size, air being passed through them by means of fans. The power, air flow and pressures were measured and the desired formulae developed.

Experiments were also performed in a model tunnel constructed in an experimental mine, to demonstrate the novel ventilation system and to determine whether upward or downward air movements would provide the more efficient ventilation, and it was proved that admitting air from the lower duct and exhausting through the upper one was the better system.

The ventilation system of the Holland Tunnel has proven

highly successful during sixteen months of operation, in which time over 11,000,000 motor vehicles used the tunnel. Other tunnels have recently been built and still others are now under construction in various countries.

With the successful ventilation of long vehicular tunnels (of large capacity and used by motor vehicles) a demonstrated fact, much popular and engineering interest is being shown in vehicular tunnel construction.

Experiments on the Permeability to Heat of Reinforced Concrete Chimneys by Prof. Otto Graf.

In the present work firstly the temperature drop and the strains caused by it in a chimney wall are discussed and the conditions for the construction of chimney walls are derived from it. In order to ascertain the actual conditions, Prof. Dr. Ing. Morsch suggested the execution of trials. In these trials the effect of different ways of insulating the chimneys as well as the influence of the velocity and of the temperature of gases had to be investigated. The results are summarized and the deficiencies of knowledge are discussed.

Ventilation of the Railway Tunnels, by Ing. Cav. Uff. Efisio Vodret.

The report in general treats the question of ventilation of underground railways, but in particular, the ventilation of tunnels in operation, *i. e.,* regarding the important problem, which especially insofar as those lines worked by steam are concerned, has always been one of the fundamental points of railway service, but in regard to trains and the personnel who stop or pass through the tunnels for line service.

It reviews the main points of the problem from a theoretical and practical side; it mentions the various systems of artificial ventilation and pauses to explain the system that the Italian engineer, Marco Saccardo, invented, tried and developed thoroughly, solving and in a simple way, the important problem.

The report shows how the idea first came to the inventor, and how it was perfectly realized mechanically, so that the *Saccardo System* is actually the best for artificial ventilation of the longest railway tunnels. To the first plant, tried with reserved confidence, great plans followed with the same fundamental principle, improving only the executive means with the developments and results of mechanical engineering.

The problem originated several years ago (1889) but yet it is, and will remain of actuality till the railroad systems in the world are improved and completed.

The report aims to keep alive the important questions showing the old and new elements interesting to the various sides of it. The question is treated:

1. With regard to the building of tunnels.
2. With regard to their operation.

The second part is that which has had the greater and more detailed development.

After mentioning the various phases of the problem, the report illustrates the *Saccardo System,* its theory, and the main points which characterize the principle, *i. e.*:

1. The *Saccardo System* is superior inasmuch as it provides for the general ventilation of the tunnel, thus improving the breathing conditions of the train staff, and line personnel.

2. The *Saccardo System* does not require the building of special wells or subterranean ways which are sometimes noxious; results can be obtained from an inexpensive plant and it can be applied to tunnels already in operation without hindering the traffic.

3. An injecting Saccardo apparatus is always efficacious in attenuating the effect produced by the gas left by the engine, not only insofar as the steam caused by the train in ascent, is concerned, but also when suppressing or weakening the steam itself.

4. Ventilation by the *Saccardo System* can be regulated according to the need, and with the greatest effect.

5. Lastly the *Saccardo System* constitutes a successful, practical and economical solution of the problem of ventilation of underground railways, and it is convenient to take same into consideration in the various cases.

The report, furthermore, presents an interesting comparison between the pneumatic efficiency of the converging injecting Saccardo apparatus and that for the direct emission of air into tunnels by tubes.

The report finally, in short, reviews the various plants made of the system from the beginning to the present time, especially mentioning the most up-to-date, in Italy.

The report is illustrated by drawings and photographs.

Train Heating on the Electrified Lines, by Ing. Cav. Severo Bissone.

Allusion is made to the usual winter heating system on electrically driven trains, by means of boiler wagons for the production of necessary steam.

Mention is also made of an experiment which has now been abandoned owing to the unsatisfactory results obtained, with electrical boilers mounted on several locomotives of the three-phase system, 3600 volts and 16.7 periods.

Water contained in said boilers of considerably small volume acted as a conductor and connected directly with the line phases and the boiler body connected with the third phase (ground).

Having mentioned that the heating by means of electrical radiators is adopted on the Italian electrical railways only on a small part of material in service on lines with 650 volts continuous current, and it is noted finally that recent experiments have been made whose results have encouraged their development, with naphtha fuel boilers mounted directly on locomotives and consequently to be taken care of the locomotive drivers and evidently with considerable economy.

In addition, the following papers referring directly or indirectly to heating and ventilation were submitted:

On the Cooling of an Under-Ground Hot Piping, by S. Nukiyama.

On the Process of Drying Cocoons at a High Temperature, by G. Imai.

On the Thermal Flexure of a Flat Slab Heated Uniformly on One Surface, by N. Yamaguchi.

The Trend of Modern Heating in the British Isles, by J. R. Preston.

New Problems in the Heating Industry, by C. W. Brabbee.

Development of Water Tube Boilers in Japan, with Special Reference to "Takuma" Boiler, by M. Kamo.

New Earthquake-Proof Socket Joint of Cast Iron Pipe for Water Main, by Z. Takenaka.

Although the Congress closed officially on November 7, 1929, events were scheduled up to November 22, 1929. These included excursion trips and inspection tours. At the conclusion of the Congress many of the American guests continued their trip around the world visiting China, India, Africa and European countries.

Niagara Blower Office Opened

W. G. R. Braemer of Philadelphia has opened an office in the La Fayette Bldg., for the Niagara Blower Co., Buffalo, N. Y.

New Address for Ilg

J. M. Frank, vice-president of Ilg Electric Ventilating Co., Chicago, Ill., manufacturers of fans, blowers and unit heaters, has announced that the Chicago sales office is now located at 182 North La Salle St.

Local Chapter Reports

Cleveland

December 13, 1929. The members and guests of the Cleveland Chapter met in the rooms of the Cleveland Engineering Society on December 13, for their regular monthly meeting.

At this meeting, H. M. Nobis was selected to represent the Chapter as a member of the Nominating Committee at the Annual Meeting of the Society in Philadelphia in January.

An interesting paper was read by H. M. Nobis, heating and ventilating engineer, describing a unique heating and ventilating system in the Allen School, Elyria, O. Mr. Nobis illustrated his talk with photographs and drawings, which was followed by a most interesting discussion, and later by adjournment.

Illinois

January 13, 1930. E. P. Heckel, the principal speaker of the evening, described the ventilating, air conditioning and refrigerating equipment at the Chicago Stadium. There are 16,300 permanent and 6,542 temporary seats; an oval floor 145x245; 85,000 ft. of 1¼ in. brine piping through which 1400 gallons of brine circulate per minute. Seven hours to cool and 4½ hrs. to spray the floor are required to make a sheet of ice ½ in. thick. A lively discussion of the subject followed.

The proposed reduction in the personnel of the Division of Heating, Ventilating and Industrial Sanitation in the City Hall, and in particular, the reduction of engineers from nine to three, was discussed by H. G. Thomas. He drew attention to the fact that the City Council had voted to retain the nine; probably largely as a result of letters of protest from this Chapter to the Mayor, Aldermen and others and the personal solicitation by Messrs. Thomas, DeLand and Heckel. Mr. Thomas read letters from the Chapter and a response from the Mayor and pictured the inevitable slowing up in the industry which would have resulted if the department were unable to make inspections promptly.

Mr. Aeberly discussed the need of better cooperation in the industry and the need of a board or commission to represent the architects, consulting engineers, ventilating contractors, steam fitting contractors, and the Society, and any other group that may be responsible for construction work. He then proposed the following resolution: "That a committee of three or five be appointed to investigate the need of a Better Relations Commission and to report back at the next regular meeting the feasibility of such plan and if feasible, a brief outline of such a Commission's duty as well as its scope and powers." This motion was seconded by Mr. Milliken and was unanimously passed.

Kansas City

December 9, 1929. The December meeting of the Kansas City Chapter was held at the Ambassador Hotel, on December 9, with an attendance of 41 members and guests.

The meeting was called to order by President Clegg, who presided over the meeting, followed with the reading of several reports given by the various Committee Chairmen.

The principal subject of the evening was the Effect of Air Space and Surface Coefficients on Heat Transmission, which was presented by Prof. F. B. Rowley, University of Minnesota. Professor Rowley gave a very interesting lecture, using lantern slides to illustrate the tests, which were conducted at the Laboratory of the University. He further elaborated upon past experiments, giving the value of air spaces of different widths, as regards their insulation value.

This paper was an advance discussion of the same subject which Professor Rowley will present at the Annual Meeting of the Society in Philadelphia, January 28 to 31.

An interesting discussion followed concerning various surface coefficients and the fact that in insulation over 3 in. was of no additional value and might prove detrimental due to additional air circulation in large air spaces.

Economy of insulation in houses was then reviewed by Amdi Worm, who also discussed a recent paper appearing in the JOURNAL of the Society, which attempted to rate various insulations.

This meeting was one of the largest meetings held by the Chapter, and was attended by as many guests as there were members.

Following Mr. Worm's discussion, the meeting was adjourned.

Michigan

January 13, 1930. This meeting was held at the Cadillac Athletic Club, Detroit, with 94 members and guests in attendance.

One of the speakers of the evening was Soren Thorensen, Engineer of Designs of the Detroit and Canada tunnel, who was introduced by Col. B. R. Value, Executive Engineer. Mr. Thorensen used slides to illustrate his lecture, which covered in detail the design of this tunnel. It is interesting to note that for ventilating the tunnel, the new air is admitted near the floor and is exhausted at the top of the tunnel. The air supply is based on 42 air changes per hour.

The other speaker was Prof. W. D. Henderson, Director of the Extension Division of the University of Michigan, whose subject was, What the Machine Age is doing for Us and to Us. Professor Henderson declared that the time rate of change at the present is the greatest in all history and that the time rate of change during the next 25 years will influence the world for the next one thousand years.

J. Whelan and E. E. Dubry outlined the February meeting which is to be held at the Delray Plant of the Detroit Edison Co. J. R. McColl, A. Rowe and J. F. McIntire spoke in behalf of the Annual Meeting in Philadelphia, and Cecil Farrar, Chairman of the National Membership Committee, made a spirited plea for new members.

New York

January 20, 1930. A paper dealing with Steam Service from Central Stations in New York City, prepared by William J. Baldwin, Jr., Chief Engineer of the New York Steam Corp., was presented at this meeting.

L. S. Phillips, Assistant Manager of Commercial Relations, read the paper, and A. R. Mumford, Chief of the Technical Department, in charge of plant operations, presented stereopticon slides illustrating the address. After outlining the history of the company and its development as a public utility in supplying steam for a variety of large structures, Mr. Phillips said that the capital investment of the company will exceed $45,000,000.00 before the close of this year. Last year 20,000 feet of new mains were laid, generally five feet below the street level, but it is sometimes necessary to go down 18 to 20 ft. The largest mains are 24 in. with extra heavy steel flanges welded to the pipe, the lengths being 24 ft. 3 in. of steel pipe, bolted together and tested to 250 lb. per square inch hydraulic pressure. Corrugated expansion joints are used, two joints being located on either side of an anchored base. Special pipe conduits with 6 in. concrete reinforced base properly drained and filled with mineral wool enclose the mains. The mains are covered with 2 in. asbestos sponge covering. The subject of superheated steam has been much discussed and considered,

but has not been adopted by this utility. The loss of steam by condensation in the mains is comparatively small and there are other distribution steam losses through leaking traps, etc., but this is not large or sufficient to interfere with the economical distribution of the service as a whole. Metering is an essential part of the distribution system and one which is handled by an experienced staff and operating force. A steam flow meter with a variable orifice is the principal type used by this utility in metering steam to consumers. Some condensation meters are used, particularly for very small consumers, where satisfactory records are not obtainable with the smallest size flow meter manufactured. The substantial accuracy of the meters is guaranteed by the utility to within 3 per cent and meters are tested under the supervision of the Public Service Commission in accordance with the terms of the service agreement under which steam is supplied.

The regulation of the steam pressure is part of the distribution work, and for this purpose service reducing valves are supplied to reduce from street pressure to the lower fixed pressures required by the consumers. Secondary reductions in pressure for heating and other exceedingly low pressure requirements are handled by the consumers by installing secondary reducing valves.

The paper also dealt in detail with the costs of the service and the company's policy with regard to the engineering service rendered. It was stated that from the engineering viewpoint, the stations in which the steam is produced are the vitals of this utility. Steam is its product made from water and fuel. It is, however, more than *boiled water*. Over one-half of the entire capital is invested in boiler plants.

The attendance at this meeting was 104, including about 30 guests from the New York Steam Corporation.

Philadelphia

January 9, 1930. The regular meeting of the Philadelphia Chapter was devoted primarily to the election of officers, and the transaction of numerous items of business in connection with the routine affairs of the Chapter.

Thornton Lewis, National President, outlined briefly the numerous papers scheduled for the 36th Annual Meeting of the Society, and made an earnest plea for a big increase in the local Chapter membership. Each member of the local Chapter was asked to make a special effort to obtain at least one new member before January 27, 1930. Mr. Lewis also pointed out that no engineer can be greater than the profession with which he is affiliated, nor of the engineering society of which he is a member. He further pointed out that the influence of the Society can only be broadened by the increase in its membership, and by securing the best engineering ability available for the presentation of papers at Chapter and National meetings. A lively discussion followed, after which J. D. Cassell was elected Chairman of a special committee for increasing the membership of the Philadelphia Chapter.

H. P. Gant gave a very interesting and comprehensive report on the *International Heating and Ventilating Exposition* to be held at the Commercial Museum January 28 to 31, 1930, inclusive, in connection with the Annual Meeting of the Society, quoting figures which indicated that the Exposition would unquestionably be an outstanding success both from an engineering and financial viewpoint.

R. C. Bolsinger, General Chairman for the 1930 Annual Meeting, reported that very satisfactory progress has been made on plans for the meetings. He also promised some unusually interesting features and surprises for the Annual Dinner and Dance to be held Thursday evening, January 30, and urged all members to make reservations at the earliest possible date in order to avoid the last minute rush.

M. F. Blankin, Chairman of the Meetings Committee, W. A. Bornemann, Chairman of the Membership Committee, and C. D. Graham, Chairman of the Year Book Committee, gave brief reports on the work performed by their various committees during the past year.

H. P. Gant nominated R. C. Bolsinger as delegate, and A. J. Nesbitt as alternate, for the local representative on the National Nominating Committee of the Society. The nomination was properly seconded by J. D. Cassell, and these gentlemen were elected unanimously.

The motion being properly seconded, was carried, and the following officers were unanimously elected:

President ..... H. G. Black
Vice President ..... E. N. Sanbern
Secretary ..... L. C. Davidson
Treasurer ..... W. H. Wild

Board of Governors: A. C. Edgar, A. J. Nesbitt, M. F. Blankin.

St. Louis

January 8, 1930. The regular monthly meeting of the St. Louis Chapter of the Society was held at the Roosevelt Hotel, January 8, 1930, with President White presiding over a gathering of 22 members.

J. M. Foster, chairman for the Program Committee, reported that the unit heater division of his Committee was going to provide the program for the February meeting and that Gas Heating would be the subject for the March meeting. It was also decided that the June meeting should be given over to Ladies' Night, as has been the custom for some time.

C. A. Pickett, chairman of the Entertainment Committee, announced that there would be a dinner dance in February, which was followed by various reports from the Committee chairmen.

As a token of appreciation for his activity and loyalty to the Chapter when he was able to take part in its activities, and as an expression of sympathy for him in his present illness, the Chapter voted unanimously to send Vincent Rossman a paid bill for his dues for the ensuing year.

The Chapter, by unanimous vote, selected C. A. Pickett as representative for the Nominating Committee for the Annual Meeting at Philadelphia, and G. Myers as the alternate.

Tom Gale of the St. Louis *Heating and Piping Contractors Association* gave an interesting and instructive discourse on the Heating and Piping Contractors National Standards, illustrating his talk with several examples for figuring radiation. At the conclusion Mr. Gale was given a rising vote of thanks by the members present, after which the meeting adjourned.

The Guide 1930

The A. S. H. V. E. Guide 1930, which is the eighth annual edition and comprises nearly 1000 pages, is now available. "The most important book in its field," contains a vast amount of new and revised data gathered from a variety of sources. They are presented in readily usable form with the hope that an effective service in the advancement of the industry will be rendered.

Truth and service are supreme in the conception of this volume. It is felt that these form the lifeblood of the profession and of all industries allied to the profession. There has been no deviation from the principles upon which The Guide was founded in 1922. The makeup of The Guide 1930 has been changed radically in the hope that it will make the book easier to use. Several of the older chapters have been consolidated and an effort has been made to coordinate the various subjects treated in the 34 chapters.

All of the data have been prepared by carefully selected Engineers, each pre-eminent in his specialty, and have been summarized and tabulated for ready reference. Several new chapters appear in this edition of The Guide, including chapters on coal, electrical heating, physical units and tables, symbols and abbreviations, specification items, and special heating and ventilation applications. Heating and ventilation have been treated

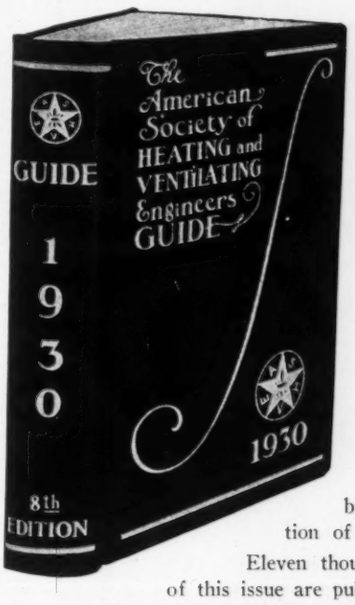
together rather than separately, for the reason that they are so related and interdependent that no definite division can be drawn.

All other chapters have been revised in accordance with current knowledge and practice. The heat transmission tables of Chapter 2, Heat Losses from Buildings, have been enlarged and rearranged to include many common types of construction, and all the tables have been recomputed using a new conductance factor for air spaces. A discussion of the economic value of insulations has been added to this chapter, in addition to data regarding condensation on interior building surfaces.

Chapter 3, Standards of Ventilation, has been condensed, and important new data added. Chapter 4, Systems of Ventilation, includes a discussion of natural ventilating systems, gravity ventilating systems and mechanical ventilating systems. Chapters on both gravity and fan pressure warm air systems are included in THE GUIDE, and data on a fan-furnace system for residences and other small buildings incorporated.

The chapter on Air Conditioning has been especially improved, and considerable available information and data on this subject included. A particularly valuable chapter of THE GUIDE 1930 is that relating to piping for steam heating systems, which has been rearranged so that the pipe size data are much more readily understood and applied. The usefulness of the chapter on Drying has also been greatly increased by the addition of considerable new data. The new chapter on Special Heating and Ventilating Applications includes reference to a number of important special phases of this subject, including greenhouse heating, dairy barn ventilation, government buildings, heat exchangers, cold storage warehouses, zoning, and air conditioning of libraries.

The arrangement and indexing of THE GUIDE 1930 have had special study so that the book may be used with maximum effectiveness. A start has been made this year to supplement the Technical Data with Catalog Data in such form that it "continues the story." Many manufacturers have prepared copy which gives specific instructions to amplify the general rules which appear in the text. An endeavor has been made not so much to enlarge the copy as to improve the content and format.



In the Catalog Data section, the manufacturer has been invited to present the complete story of his products so that the user will have before him concise and complete information which will enable him to make the best selection which modern equipment affords. It has been the policy that the manufacturers' data presented in the Catalog Data section be useful and serviceable,—not a mere collection of claims, high-sounding catch phrases or superlatives, but an informative presentation of helpful information.

Eleven thousand, two hundred copies of this issue are published and it is offered to the profession and the industry at large with the hope that it will render an unique service as the standard reference work on heating and ventilating.

This work was produced under the direction of the *Guide Publication Committee*, S. R. Lewis, *Chairman*, C. V. Haynes, W. H. Carrier and J. F. McIntire, *Vice-Chairmen*, and P. D. Close, *Technical Secretary*.

The *Guide Publication Committee* will greatly appreciate suggestions for improvement of the next edition, which suggestions should be forwarded to the headquarters office of the SOCIETY, 29 West 39th Street, New York, N. Y.

THE GUIDE 1930 has a semi-flexible, blue binding stamped in gold, and may be obtained for $5.00 per copy.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society: All applications for membership are to be sent to the Secretary and the name of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 54 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by March 1, 1930, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary, immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ATHERTON, F. R., American Radiator Co., Detroit, Mich.	G. H. Giguere C. W. Farrar	W. G. Boales H. C. Kappler
BRAATZ, CHESTER JOHNSON, Inst., Steam & Gas Engrg., Univ. of Wisconsin, Madison, Wis.	G. L. Larson F. C. Houghten	W. M. Richtmann F. G. Weimer
BROWN, TOM, B. F. Reyolds & Co., Detroit, Mich.	G. H. Giguere L. L. Smith	H. L. Walton L. L. McConachie
BUCKLEY, MARTIN B., New Business Manager, Kansas City Gas Co., Kansas City, Mo.	C. A. Weiss J. H. Kitchen	W. E. Gillham N. W. Downes

Byrnes, Emmet F., Manager, H. Kelly & Co., Detroit, Mich.	G. H. Giguere H. M. Stephen	L. L. Smith W. A. Blessed
Cantwell, John DeMuth, Engr., Trane Co., and A. Harvey's Sons Mfg. Co., Detroit, Mich.	G. H. Giguere W. G. Boales	R. S. M. Wilde N. B. Hubbard
Conover, Edmond W., Research Engr., Detroit Steel Products Co., Detroit, Mich.	G. H. Giguere W. C. Randall	H. C. Kappler J. E. Degan
Cooper, Frederick D., Excelso Products Corp., Buffalo, N. Y.	G. H. Giguere R. K. Milward	E. E. Dubry L. L. McConachie
Cossaboom, George Carlton, Weston & Ellington, Detroit, Mich.	G. H. Giguere J. L. Fuller	W. J. Whelan E. H. Clark
Cowan, Edward, Jr., The Cowan Co., Detroit, Mich.	G. H. Giguere W. A. Blessed	H. C. Kappler L. L. Smith
Cundall, Lincoln Atwood, Estimator & Asst. to Pres., L. A. Harding Constr. Corp., Buffalo, N. Y.	L. A. Harding C. A. Booth	O. K. Dyer R. T. Thornton
Decker, Emil J., Automatic Coal Burner Corp., West New York, N. J.	C. W. Hall Noel Cunningham	F. C. Kittle D. H. Faulkner
Faltenbacher, Harry John, Plbg. & Htg. Contractor, 235 E. Wister St., Philadelphia, Pa.	W. H. Wild R. C. Bolsinger	Lee Nusbaum W. G. Culbert
Fisher, Albert E., Sales Engr., Kerr Mchy. Corp., Detroit, Mich.	G. H. Giguere W. J. Whelan	W. A. Blessed H. C. Kappler
Gallarno, Charles A., Secy., Treas., The Donald Miller Co., Detroit, Mich.	G. H. Giguere H. M. Stephen	W. A. Blessed W. J. Whelan
Giddey, York, R. F., Vice-Pres. & Genl. Mgr., Bryan Boiler Sales, Inc., Detroit, Mich.	G. H. Giguere W. G. Boales	R. R. Peckham N. B. Hubbard
Guilford, Frederick William, A. Harvey's Sons Mfg. Co., Detroit, Mich.	G. H. Giguere J. L. Fuller	R. K. Milward J. E. Degan
Harney, Francis W., Dir. of Engrg. Research, Upson Co., Lockport, N. Y.	F. H. Burke	L. A. Harding H. B. Roarke
Harrison, Charles George, Crane Co., Detroit, Mich.	G. H. Giguere H. M. Stephen	W. A. Blessed H. C. Kappler
Hassberger, H. C., Lorne Plbg. & Htg. Co., Detroit, Mich.	G. H. Giguere H. M. Stephen	J. C. Degan E. H. Clark
Hodges, George S., Jr., Detroit Stoker Co., Detroit, Mich.	G. H. Giguere J. L. Fuller	N. B. Hubbard E. H. Clark
Jennings, Warren G., Res. Vice-Pres. & Branch Mgr., Minneapolis-Honeywell Regulator Co., Chicago, Ill.	H. G. Thomas C. W. Johnson	C. W. DeLand J. J. Aeberly
Kelley, Arthur D., Jr., Sales Engr., York Htg. & Vtg. Corp., New York, N. Y.	H. B. Hedges H. P. Gant	W. H. Price C. D. Graham
King, David J., American Blower Corp., Detroit, Mich.	G. H. Giguere H. E. Paetz	N. B. Hubbard H. M. Stephen
Meyer, Charles L., L. J. Wing Mfg. Co., New York, N. Y.		
Molinet, Ignacio S., Carrier Engrg. Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	M. S. Smith L. L. Lewis
Moller, Robert Christian, Carrier-Lyle Corp., Boston, Mass.	G. G. Schmidt V. S. Day	A. E. Stacey, Jr. C. D. Park
Nagelson, Lou M., Western Sales Mgr., Buckeye Blower Co., Columbus, Ohio.	W. I. Lawson S. E. Fenstermaker	C. N. Warren R. W. Noland
Ohnell, Ernest, Jr., Sales Mgr., Midwest Mfg. Co., Bradford, Pa.	John Hellstrom E. B. Cresap	W. M. Reed J. F. Reed
Ormsby, H. Kingsley, Jr., Hoffman Specialty Co., Waterbury, Conn. (*Advancement*)	C. W. Stewart J. K. Peacock	W. K. Simpson M. C. Gillett
Park, John F., Carrier Engrg. Corp., Newark, N. J.	W. G. Hillen Herbert Coward	W. H. Carrier H. Worsham
Parrock, Edwin J., American Blower Corp., Detroit, Mich.	G. H. Giguere H. C. Kappler	W. A. Blessed H. M. Stephen
Peacock, Herbert, Dist. Mgr., Carrier Engrg. Corp., Detroit, Mich.	G. H. Giguere L. L. Smith	W. A. Blessed H. C. Kappler
Peterson, Sterling Donald, Seattle Repr., Johnson Service Co., Milwaukee, Wis.	W. W. Cox W. E. Beggs	Marius Anderson P. M. O'Connell
Philip, William, Asst. Htg. Engr., McKellar & Blackhall, Toronto, Ont., Canada.	W. R. Blackhall W. G. Sheppard	H. S. Moore L. A. Hood
Poe, Thomas O., Pres., Poe Co., Detroit, Mich.	G. H. Giguere H. C. Kappler	C. W. Farrar W. G. Boales
Purdy, Ellwood D., Pres., Hardinge-Detroit Corp., Detroit, Mich.	G. H. Giguere J. L. Fuller	W. J. Whelan W. A. Blessed
Rachal, John Meredith, Carrier Engrg. Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	F. P. Anderson H. Worsham

RANDALL, HERBERT N., Sales Engr., Warren Webster & Co., Detroit, Mich.	G. H. Giguere R. K. Milward	H. C. Kappler W. G. Boales
RANDALL, ROBERT D., D. T. Randall & Co., Detroit, Mich.	G. H. Giguere H. M. Stephen	W. A. Blessed H. C. Kappler
READER, JOSEPH THOMAS, Vice-Pres., Kerr Mchy. Corp., Detroit, Mich.	G. H. Giguere H. L. Walton	W. A. Blessed H. C. Kappler
SANDS, JOHN S., Westinghouse Elec. & Mfg. Co., Philadelphia, Pa.	H. M. Patrick L. C. Davidson	E. N. Sanbern R. F. Hunger
SANFORD, STERLING S., Htg. Engr., The Detroit Edison Co., Detroit, Mich.	G. H. Giguere J. H. Walker	N. W. Calvert E. E. Dubry
SHAVER, ELMER W., Sales Engr., Coon-DeVisser Co., Detroit, Mich.	G. H. Giguere H. C. Kappler	J. L. Fuller W. A. Blessed
SHEA, MICHAEL B., American Radiator Co., Detroit, Mich.	G. H. Giguere H. R. Linn	H. M. Armstrong W. A. Rowe
SKILLIN, N. MORTON, Dist. Sales Mgr., Cohoes Rolling Mill Co., Detroit, Mich.	G. H. Giguere H. C. Kappler	W. G. Boales H. M. Stephen
SMITH, C. WARNER, Dist. Mgr., Ilg Electric Ventilating Co., Philadelphia, Pa.	J. M. Frank B. L. Casey	B. W. Wilson E. N. Sanbern
STEWART, EARL ALBERT, Estimator & Engr., McVehil Plbg. Co., Washington, Pa.	C. W. Wheeler H. E. Digby	E. C. Evans E. W. McVehil
SWARTZ, HENRY A., Sales Engr., U. S. Radiator Corp., Detroit, Mich.	G. H. Giguere J. F. McIntire	E. E. McNair W. A. Blessed
SYLVAN, STIG G., Research Engr., Midwest Mfg. Co., Bradford, Pa.	John Hellstrom E. B. Cresap	W. M. Reed H. C. Murphy
TRETHAWAY, RAY S., Sales Engr., Chicago Pump Co., Chicago, Ill.	G. H. Giguere H. C. Kappler	E. H. Clark J. L. Fuller
WALTER, WILLIAM, York Htg. & Vtg. Corp., New York, N. Y.	H. B. Hedges H. P. Gant	W. H. Price C. D. Graham
WARREN, WALTER JOSEPH, Smith, Hinchman & Grylls, Detroit, Mich.	G. H. Giguere L. L. Smith	H. M. Stephen W. A. Blessed
WILLETT, WILLIAM McN., York Htg. & Vtg. Corp., Philadelphia, Pa	F. P. Anderson Thornton Lewis	W. F. Noll J. S. Jung

Candidates Elected February, 1930

In past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

BUCKLEY, JOSEPH L., Carrier Engineering Corp., Philadelphia, Pa.

DAWSON, THOMAS L., Thos. L. Dawson Co., Kansas City, Mo.

FARLEY, W. F., Mgr., Heating Dept., Crane Co., New York, N. Y.

GRAVES, CLARENCE C., Secy.-Treas., Graves & Graves, Inc., 3047 Sheffield Ave., Chicago, Ill. (*Advancement.*)

ISRAEL, CHARLES HENRY, JR., Salesman, York Heating & Ventilating Corp., Philadelphia, Pa.

MANN, ARTHUR R., Owner, Mann & Co., Archts. & Engrs., Hutchinson, Kans.

NESSI, ANDRI, Nessi Freres & Cie, Paris, France.

SCHAFER, HARRY C., Dist. Sales Mgr., B-Line Gas Boiler Co., Cleveland, Ohio; Skinner Bros. Mfg. Co., Inc., St. Louis, Mo.; Mears-Kane-Ofeldt, Inc., Philadelphia, Pa.

WALLACE, R. ROBERT, B. F. Sturtevant Co., Chicago, Ill.

ASSOCIATES

ANDERS, GEORGE W., Pres., Sterling Engrg. Co., Milwaukee, Wis.

AXTHELM, FRED G., 648 N. Forest Ave., Webster Groves, Mo.

BRAKENRIDGE, CHARLES EARL, Anthracite Coal Service, Chicago, Ill.

CUNNINGHAM, TOM M., Branch Mgr., Carrier Engrg. Corp., Dallas, Tex.

DEPPE, FRED W., Dist. Sales Repr., 1823 Railway Exchange Bldg., St. Louis, Mo.

GERSTUNG, G. WALTER, Multicell Radiator Corp., Buffalo, N. Y.

GLANZ, EDWARD, Glanz & Killian Co., Detroit, Mich.

HARRIGAN, HOWARD H., Consolidated Ashcroft Hancock Co., Detroit, Mich.

HARRINGTON, ELLIOTT, Engrg. Dep., General Electric Co., Schenectady, N. Y.

HILL, CHARLES E., Hoffman Specialty Co., Inc., Waterbury, Conn.

LILLARD, WALTER S., Plbg. & Htg. Contractor, Peru, Ind.

MANNEN, HENRY C., JR., Walworth Co., Detroit, Mich.

MICHIE, D. FRASER, Crane, Ltd., Winnipeg, Manitoba, Canada.

MOFFITT, ROY M., Branch Manager, L. J. Mueller Furnace Co., Milwaukee, Wis.

MONROE, RAYMOND R., Nash Engrg. Co., South Norwalk, Conn.

RICHARDSON, HENRY THOMAS, Vice-Pres., Richardson & Boynton Co., New York, N. Y.

TURLAND, CHARLES HENRY, Asst. Mgr. & Vice-Pres., Acme Fan & Blower Co., Ltd., Winnipeg, Manitoba, Canada.

WINQUIST, WALTER J., Clark, MacMullen & Riley, New York, N. Y.

WINQUIST, WILLIAM M., Sales Engr., Silent Automatic Sales Corp., New York, N. Y.

MUELLER, HARALD CHARLES, Sales Engr., The Powers Regulator Co., Chicago, Ill.

PRICE, LEWIS CAIRL, L. C. Price & Co., 136 Federal St., Boston, Mass.

SZOMBATHY, LOUIS R., Pres., Ferguson Sheet Metal Wks., Inc., Ferguson, Mo.

JUNIORS

ASBURY, NORMAN B. G., Sales Engr., York Heating & Ventilating Corp., Philadelphia, Pa.

BARNARD, MILTON E., Sales Engr., York Heating & Ventilating Corp., Philadelphia, Pa.

EASTMAN, CARL B., Dist. Mgr., C. A. Dunham Co., Philadelphia, Pa.

HARRIGAN, EDWARD R., Harrigan & Reid Co., Detroit, Mich.

KEITH, LEWIS HAYDEN, Sales Engr., York Heating & Ventilating Corp., Philadelphia, Pa.

SNYDER, ALLEN K., Carrier Engrg. Corp., Newark, N. J.

TAYLOR, KENNETH A., York Heating & Ventilating Corp., Philadelphia, Pa.

WACHS, LOUIS J., Carrier Engrg. Corp., Newark, N. J.

Heating · Piping
and Air Conditioning

EDITORIALS

Giving Thought to Piping

It pays to have the various piping systems in the big industrial plants designed, installed and maintained according to best engineering principles. Because of its direct relation to fuel and pumping costs as well as to heating, power and process efficiency, piping deserves rank as one of the important phases of plant operation and maintenance. It pays, as we say, therefore, to have it right.

HEATING, PIPING AND AIR CONDITIONING made piping one of its major editorial features because investigation developed the fact that it was a leading engineering problem for one or more men in the larger industrial plants and it was not being given first attention in the technical press. So, on the basis that industrial piping should be right, that to have it right required engineering knowledge and study, one reason that this magazine is published is to give piping the voice and attention it deserves.

We recall two interesting experiences in the course of investigating the importance of piping in the minds of operating engineers in various plants throughout industry. These engineers were practically unanimous in their statements that piping was a major part of plant operation and maintenance, but there were the few who felt that a pipe line was just a proposition of connecting the requisite number of lengths of pipe and that engineering knowledge and study were not so essential in doing that as a strong back and the right tools. The experiences to which we refer were such exceptions.

One was the manager of a good-sized pulp and paper mill (no one had an engineering title there). This manager politely ridiculed the idea that his steam lines, acid lines, etc., represented a problem of engineering design, saying that when it was necessary to change a line, run a new one or extend an old one, he had a couple of handymen who managed to get it from point to point. In contrast, he explained how competitive pulp and paper mills maintained engineering departments and, before doing the same kind of job, would indulge in a lot of figuring, drawing, and what not. Then, he hesitated, looked a bit thoughtful and said, "You know, we never gave much of any thought to piping around here. Maybe we should give more."

They should indeed, experience would indicate. The proposition of running a pipe line from one point to another involves a nice balance of original cost and operating and maintenance expense.

The other was the supervising engineer of an eastern textile mill. He had difficulty in getting serious about what piping meant to his plant because for a year or more business in his company's end of the textile industry had been pretty bad, and his instructions were to cut operating expenses to a minimum and to plan no new changes or extensions. Therefore, as he said, they were getting along as well as possible with existing equipment, some of which was obsolete, some of which was in need of major repairs, etc. But then he, too, rallied us back to our championship of right piping by remarking, "It takes more engineering to economize than it does to go ahead on a liberal budget."

Which is also right, and to the point, we say. Economy demands correct piping design, installation and maintenance whether money be plentiful or scarce.

Quite a While?

The *New Yorker,* commenting on the method proposed by Dr. Frazer of Johns Hopkins for eliminating carbon monoxide from exhaust gases discharged by automobiles, winds up with a statement to the effect that although we know CO to be injurious, and in large quantities, deadly, "we expect to be breathing monoxide for quite a while yet, . . ."

Mankind tolerates numerous conditions, detrimental to health and comfort, if not hazardous to life—conditions which can be corrected by the installation of proper air conditioning equipment. Summer-time finds us in an atmosphere which is often depressing, hot, and acts to reduce efficiency. In winter, we put up with a dangerously-dry condition, conducive to colds and other illness. Workers in many industrial plants are exposed to large quantities of dust, extremely dangerous to human life, as pointed out in an editorial in the January number of this publication.

Engineers, owners of office buildings, men in control of industrial plants, all know the advantages to be gained by proper conditioning of the air—advantages not only to human welfare, but to property values, to plant and office efficiency, to profits. Air conditioning has made rapid strides within the past few years. But, when one balances the advantages, the health and comfort and efficiency, resulting from the maintenance of proper temperature, humidities, and air motion, one wonders why every plant, every office building, every hotel and every structure is not properly air conditioned.

City Noise

Considerable comment has appeared lately, in the daily and engineering press, relative to the action of the president of a New York concern in writing letters of apology to tenants in buildings adjacent to his company's headquarters, which were in the process of construction, regretting the inconvenience and noise caused them, and requesting their forbearance for a few more weeks. If every concern causing noise, necessary, perhaps, but none the less irritating, were to send out such apologies, the letters themselves would soon become as much of a nuisance as the sound-waves.

Thomas Alva Edison, deaf himself, predicts that evolution will cause man to lose his hearing in the not-distant future, because of the volume of noise to which we are constantly subjected.

Much can be done to reduce noise, which appears to be inevitable to city life, by insulation for sound isolation, both at the "sending" and "receiving" ends. Many engineers predict the window-less skyscraper (properly air conditioned) as a solution of the problem. The development of machinery imposes a responsibility on society; we must keep ahead of our inventions. To maintain our health, we must reduce city noise.

Trade Literature

There is such a great quantity of trade literature published by manufacturers of heating, piping and air conditioning equipment today that many engineers in these fields are prone to neglect reading the catalogs, booklets, pamphlets and circulars that come to their attention each day. It is a problem to know what part of this material to file, and what part to discard. Many solve the problem by discarding everything; others use their files as sort of "glorified" waste-baskets.

Much valuable information is contained in these trade publications; information which business men and engineers cannot afford to neglect. We know of one very successful man in the confectionery industry; one of his outstanding characteristics was his habit of looking at every page of every piece of trade literature. He did not read every word necessarily, but he had obtained the ability of picking out the important points on any printed page and remembering them.

He died leaving a large fortune; we think much of his success was due to his intimate knowledge of the products advertised, and the information contained, in the trade literature.

Balancing Costs

Recent experiments at the University of Illinois Engineering Experiment Station on the re-alignment and removal of obstructions from underground air passageways for ventilating show that considerable economies can be effected by providing such air tunnels with smooth sides and making them as free of obstructions as possible. The power consumption was decreased enough to give substantial returns on the cost of improving several test sections.

If industry is to save on first cost, it must expect to pay proportionally higher operating charges. Often the extra operating cost is in excess of what it should be; when this is the case, it is time to increase first cost in order to reduce the excessive operating charges.

This situation applies directly to almost any installation of air conditioning, heating, or piping equipment. It is a factor that must be given careful consideration by all engineers.

New Blood

In another month or two the annual pilgrimage of representatives of industrial concerns to the engineering and technical schools of the country will begin. Seniors in all curricula will be interviewed for this or that position with this or that company.

The concerns who regularly send interviewers to our technical institutions range from the small firm to the large corporation with its many subsidiaries and its staff of personnel men devoting all its time to selection of new men. One characteristic, however, distinguishes them all.

It is always the progressive and the well-established concern that sends men out to pick college graduates for employment, and this has raised a question in our mind. Is this practice of carefully picking new blood each year one of the contributing factors to such progressiveness, or is the well-established firm the only type that can afford to train new men? We hold to the former view-point, we believe.

One large corporation knows from its records that less than a third of the men it trains will remain with it; yet the company figures it is a profitable scheme. The men who leave after their training course is finished, or within a few years, are not lost to the industry. And the prosperity of any concern depends on the prosperity of the industry to which it belongs.

An Ill Wind

Following news of the recent fire in the executive offices of the White House comes the announcement that President Hoover's business offices are to be completely air conditioned. The blaze was not, of course, the reason for the installation. When the decision to provide conditioned air for the White House was first made, it was planned to install the ductwork piecemeal, taking advantage of any absences of the President from Washington.

Such procedure would have delayed the installation for some time, no doubt, and it is probable the President would have been put to considerable annoyance during the time of installation.

The fire, and the re-building made necessary, gives the engineers a chance to provide for the air distribution apparatus. Apparently, the old saying "It's an ill wind that blows no one good" still applies.

INTERVIEWS OF INTEREST

The Engineer and the Physiologist

Dr. William H. Howell, head of the department of physiological hygiene of the School of Hygiene and Public Health, Johns Hopkins University, Baltimore, Md., discussed with some detail the more recent research relationships between physiological and engineering groups.

Earlier research preoccupations served to isolate the physiologist in the medical laboratory; and the engineer, dealing exclusively with materials and dimensions, pressures and strains, felt little need for collaborative work with the hygienist. But times have changed, and the physiologist is on constant call as referee concerning debated questions pertaining to the air environment, states Dr. Howell. The public listens, and nowadays, even the strict laboratory type of medical investigator is expected to issue rapid and ready statements on what is an optimal air condition and what "system" may be expected to attain it.

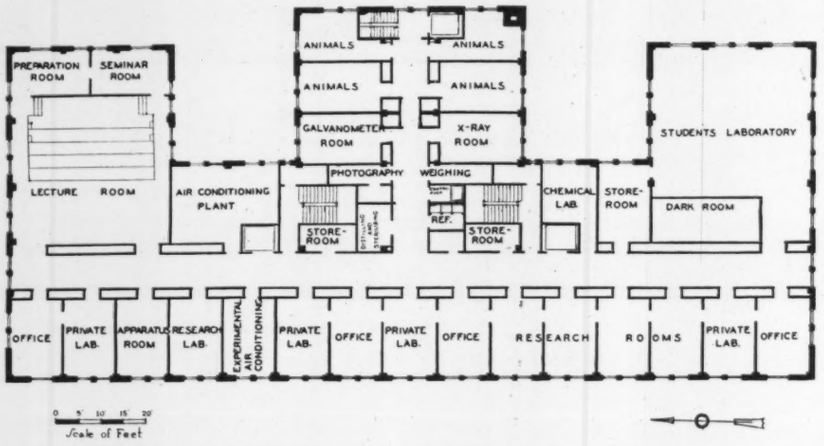


FIG. 1—SEVENTH FLOOR PLAN, SCHOOL OF HYGIENE AND PUBLIC HEALTH, JOHNS HOPKINS UNIVERSITY

The engineer finds himself in a similar quandary, for the same situation thrusts him into the role of a quasi-physiologist, hard pressed to present health equivalents for any type of air environment that may be under discussion.

"As yet, not one knows enough about the health correlations of ordinary room atmospheres to state with certainty that a given condition is in the long run in a given way injurious to health," stated Dr. Howell. He asserted that it is especially inadvisable to emphasize humidification, ozonation, freezing, filtering, baking, chemical sterilization, or de-odorization processes as single-solution panaceas for all the more common ventilation errors. We need time, first, for physiologists to work out these matters in the laboratory. The physiologist often has to rescind certain statements as new facts come in, and "fad" apparatus founded on quick generalization fares worse than this.

He went on to say that the situation with respect to industrial hygiene is much less uncertain. Here we have abrasive industries where forced exhaust of foreign particles is all the air hygiene the situation calls for. In chemical trades the toxic fume is known and can be taken care of specifically. General hygiene is much more uncertain. "What of the industries in which chronic respiratory disease prevails in high degree, but acute colds are seldom seen? Or what of highly common colds where air seems pure? Dusty trade can almost be handled by formula, but organic dusts present a different case. Professor W. Storm Van Leeuwen, of the University of Leiden, thinks to remove atmospheric allergens that are responsible for asthmas by means of special rooms externally ventilated; but that work, too, awaits a fuller check."

The School of Hygiene and Public Health is organized in full recognition of the common fields of inquiry on the technical side of industrial work. Our understanding of what constitutes normal function, and the formulation of those standards and rules of living which conduce toward normal function, must include all those more or less artificial conditions imposed upon us by our communal life, and the crowding together of individuals in industrial and social occupations of various kinds. The problems of human adaptation these things involve constitute the specific field of investigation covered by Dr. Howell's department.

Physiological Hygiene Department at Johns Hopkins

This department occupies the entire seventh floor of the School of Hygiene and Public Health. Ventilation of the building as a whole has received special treatment for the purposes it serves. Air supply with exhaust for auditorium is a separate system. Chemical hoods for laboratory desks are connected with individual fans that exhaust through the roof. Other exhaust fans serve smaller lecture rooms and student laboratories. Service rooms constitute another unit. Animal rooms are isolated from other sections of the building by means of an air lock, and draft vent flues are separate for animal rooms.

Specific studies in climatological effects are capable of being set up for group observation in a general lecture room shown on the north end of the floor plan, accommodating seventy-five students. The air conditioning apparatus located in the adjoining room can provide in this room, and in a smaller experimental air conditioning

room across the hall, a wide range of combinations of temperature, humidity and air movement.

The smaller room has been the center of the work on acclimatization which has been done in Dr. Howell's department. The air in this room is filtered, washed, cooled, humidified, heated, or made arid as set problems under investigation may require. The conditions set are capable of being maintained indefinitely without change. Regulation is precise, and continuous automatic records give the necessary check upon routine readings.

Bodily Discomfort Basis of Bad Health

"In the absence of any absolute criteria of air conditions for health," states Dr. Howell, "we can only surmise that, in the long run, bodily discomfort does become the basis of bad health. We first discover the comfort zone for the individuals to be studied, and we measure any fluctuations in efficiency that run parallel with variations in given directions of the atmospheric conditions created. Reaction time, muscle power, endurance tests, the power to sleep, quickness and validity of judgment, mental clearness, interest, precision, sense acuity—these are some of the criteria we employ.

"We believe that significant correlations can be found between skin temperatures and comfort, or the ability of the body to adapt itself efficiently to a given air environment. This year our work relates to working out these relationships. One result that does come out clearly from our work as well as the work of all observers is that the power to do muscular work falls off at high temperatures and humidity, owing to the development of fatigue.

"A single problem usually occupies a year. It takes a year to answer the type of physiological questions asked. Each problem solved suggests the next line of inquiry. One piece of work of unusual interest strongly suggests that there may be a toxic principle in expired air. Some authorities deny this, and much more work is called for; but, unfortunately, the controlling genius of this particular piece of investigation has greater interest in something else, and our work in that direction has been pursued no further.

The Common Cold

"No one knows much that is definite about the common cold. This matter is the subject of co-operative investigation going on at the present time. The study is planned for a period of five years and proceeds as a five-way inquiry: epidemiological, statistical, physiological, experimental, clinical. This work, with Dr. James A. Duoll in immediate charge, has been under way for one and one-half years. We hope it will yield positive results. The study is supported by the Chemical Foundation."

Two Barriers to Research

Dr. Howell thinks that two barriers, for the moment seemingly insuperable, obstruct any very extensive common ground of action on ventilation matters between physiologist and engineer. The physiologist has too much grubbing research to accomplish for him to indulge in any speculative effort; and the engineer has to combat too many operative problems, human and mechanical, quite apart from the scientific possibilities of his system, for him to make any very extended excursion into fields foreign to the mechanics of his profession.

Hospital Engineering Problems

"We are facing an amazing development in the hospital field," Alfred Kellogg, consulting engineer of Boston, declares. "The economic advance of this particular field is difficult to account for.

"Hospital engineering on these enterprises represents a highly varied service. It covers power, lighting, refrigeration, sanitation, or heating and ventilation; or perhaps all of these services.

"The engineering consultant who serves the hospital field in a very specific manner has to concentrate upon internal hospital service needs. Two medical clients may speak in almost interchangeable terms about their desire for constant control rooms, but one may have in mind a precision laboratory for experimental work, and the other may contemplate absolute control of atmospheric environment for premature birth wards on the basis that obtains in children's hospitals in many leading medical centers at the present time. The one may require wide ranges of temperature and humidity relationships under continuous record and control; while the other may contemplate average temperatures with more or less fluctuation of temperature and place his emphasis upon rapid and complete air change within his hospital wards.

"The hospital for contagious diseases rarely has any type of formal ventilation, but modern research work on resistance to disease points to very definite correlations with air environments and suggests manifold new engineering requirements in the future.

"Regardless of any definite specification on the part of hospital clients themselves, the engineer sees to it that intercommunication of ducts is kept down to the minimum. Some physicians like special provision for high air velocities through contagious disease hospital wards, but mostly both qualitative and quantitative matters pertaining to hospital ventilation become the prerogative of the consulting engineer.

"Usually, hospital ventilation does not mean constant change of all the air at given rates per hour. It may mean anything, from open porticos for patients to compression chambers for administration of double oxygen supply.

"Ventilation of service departments may be considered for esthetic reasons, for safety, or an efficiency staff may stipulate sanitary engineering of all types on the basis of maximum output of utility work.

"The zoning principle for heating prevails in many hospitals. Employed at first for purely economic reasons, zoning practice is now advocated by many for health.

"Work on the electrical side runs into many elaborations within the hospital. Signal systems are highly complex; shadowless light; Alpine lamp rooms; draftless outdoor conditions with humidity control to be maintained in operating rooms; refrigeration for pathological rooms; refrigeration also for organic laboratories; X-ray connections and protective devices, and a constant supply of artificial light where and when needed: all of these and more require thoughtful care in modern hospital engineering."—*S. P. M.*

PRACTICAL PIPING PROBLEMS

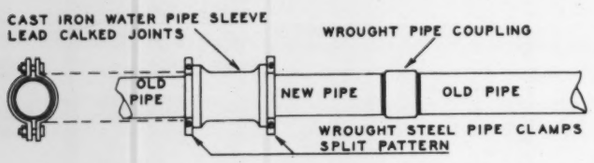
Repairing an Air Line
A Heater for Fuel Oil Piping
Hot Water Circulation

by
W. H. WILSON

Emergency Repairs to Compressed Air Line

The accompanying sketch shows repairs to a pipe line, under conditions different from the ordinary methods.

In this case, a six-inch pipe line used for transmitting compressed air was installed a foot below the surface of the ground and under several railroad tracks. Due to the heavy loaded cars passing over a joint in this line, the pipe was broken at one end of the coupling. It was essential that repairs be made, in the least possible time, as keeping the pipe



Method of Repairing Compressed Air Line

line out of service meant delay for a number of workmen.

The ground was frozen and this, with the railroad tracks on top of the pipe, meant considerable delay to dig up a full length of pipe and replace it with threaded and flanged joint.

The method used for repairs was as follows: The pipe on the side of the coupling, where the break occurred, was cut two feet from the coupling with an acetylene torch, the old coupling was removed and a new one with a piece of pipe threaded on one end and screwed in, was made up. This was connected to the end of the pipe in the ground from which the old coupling was removed. The length of this short piece of new pipe with a thread on one end was a little shorter than the distance to the point where the pipe line was cut with the torch, to allow for making up the coupling on the threaded pipe.

Before this screwed joint was assembled, a six-inch cast iron water pipe sleeve was slipped over the pipe that was cut with the torch. When the coupling and short piece were screwed up tight, the sleeve was pulled back in position over the space between the two pipe ends and this sleeve was well yarned and caulked with lead similar to the way cast iron, bell and spigot, water pipe is installed.

The air pressure was then turned on and the pipe line put in service in much less time than if a full length had been used to replace the broken one.

As an added precaution, to prevent the lead caulked joint from blowing out, a wrought pipe clamp of the split pattern was securely bolted in place as shown.

This is a quick method of repairing a pipe line in an emergency as the maintenance department of an industrial plant usually has a cast iron sleeve on hand together with acetylene cutting torches.

The pipe line mentioned was for the purpose of supplying compressed air at 100 pounds working pressure, and after these repairs were made, timbers were placed on each side of the pipe to support the railroad track and prevent the pipe breaking in a similar way in the future.

Oil Heater Made by Welding

A serviceable heater for fuel oil pipe lines can be easily made and used to an advantage, in many industrial plants, where steam is available for heating.

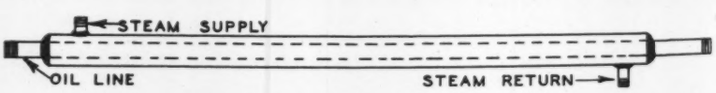
This heater is not intended to be used for the main fuel oil heater, but rather to act as a booster on long transmission lines, or in places where the oil supply or circulation lines pass through cold areas.

This heater consists of a length of pipe slipped over a section of the oil pipe line, each end welded to the line and the necessary nipples welded in the outer pipe for the steam supply and return connections. For an example, in case the size of the oil line is 1¼-in. pipe, the outer pipe could be 2-in.

A length of pipe twenty feet long can be used, without fear of expansion troubles, when steam is available at the ordinary working pressures. It would not be advisable to have any pipe joints on the oil line inside of this heater pipe.

The discharge from a steam trap can be used to supply heat for this purpose in many cases.

When the heater is used in connection with the heating



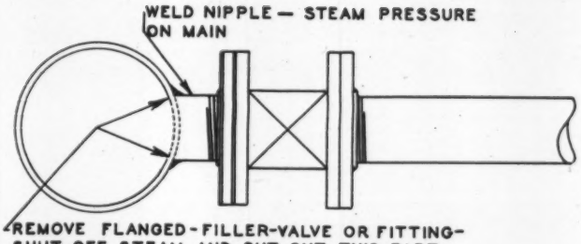
Welded Heater for a Fuel Oil Line

system for the building, it has the advantage that when steam is required for heating the building, the oil may need heating. Also, when the weather conditions do not require heat for the oil, the heating system for the building will in all probability be shut down.

A few heaters located at the places where the oil lines are affected by the cold weather will aid materially in keeping fuel oil ready for the furnace oil burners.

Installing Branch Connection to Steam Main

Alterations to piping, in many cases, require that branch connections from main supply pipe lines must be


WELD NIPPLE — STEAM PRESSURE ON MAIN
REMOVE FLANGED-FILLER-VALVE OR FITTING-SHUT OFF STEAM AND CUT OUT THIS PART OF PIPE WITH A STRAIGHT TYPE ACETYLENE CUTTING TORCH.

Method of Installing a Branch Connection on a Steam Main

installed with a minimum amount of time allowed for the supply main to be shut down.

The alterations and additions to the piping may require the work to be done in close quarters, where a definite and solid starting point is a decided advantage.

When branch lines two inches and larger are to be connected to an existing steam line, supplying steam at the ordinary pressures and in use, the following method may prove useful.

Weld a short nipple, with a companion flange on the end, to the steam main, without shutting off the steam pressure. Install a flanged valve or fitting at this point, if desired. If a valve or fitting is not needed at this point, a short section of flanged pipe or filling piece can be used. Complete the installation of the new piping required. This method will give a definite starting point, securely located in the right position.

When conditions are favorable for a shut down, shut off the steam from the main line, remove the flanged valve, fitting or filling piece, cut out for the opening in the main with a cutting torch. To prevent pieces of cut piping from falling into the interior of the main pipe, first weld a wire on the portion of the pipe to be cut out, for the purpose of withdrawing it from the opening and use a magnetized bar of steel to remove the cuttings.

Replace the flanged valve, fitting or filling piece, turn on the steam, and the branch line, as well as the rest of the piping, is ready for use.

Superheated Hot Water Circulation

Engineers in Russia have developed a scheme in hot water heating by forced circulation which has considerable to recommend it for group installations where the buildings are of reasonable height. The writer has never seen or heard of similar installations being made in this country and the idea, therefore, should prove of general interest. There is no doubt that, if the scheme works out as well as they claim it does, pipe areas and power for the circulation pump are both greatly reduced and it is unfortunate that it was not possible to see these systems under actual operating conditions of -40 F. which is the temperature for which they are designed. Anyway, here's the story.

When there are several large buildings of, say, five or six stories in height all to be heated from a central heating plant such as is illustrated in Fig. 1, the customary practice in America would be to install either a steam or forced hot water plant at the point selected and to run mains around to the various structures to be heated. In the case of forced hot water the returns would be

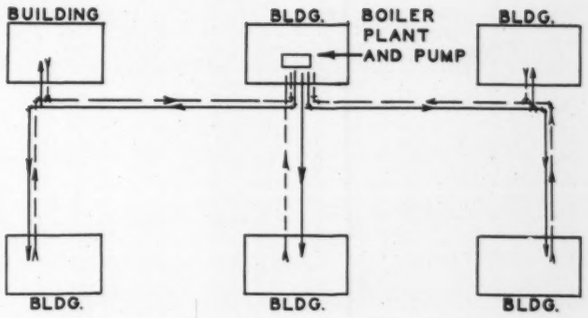


Fig. 1—Typical Plot Plan of System Supplying Six Buildings by Forced Circulation of Hot Water

carried back to the heating plant in the proper sized return mains and in the case of steam the condensation would either be carried back to the heating plant or wasted in order to economize on the installation cost.

The Way It Is Done in Russia

The scarcity of pipe in Russia and the desirability of economizing in every particular, which policy is now in full force there, makes the Russian engineer look twice before he commits himself and then he comes along with a scheme which sounds impractical on the face of it. For he suggests circulating water at 220 to 230 F from the boiler plant to the point where each building is supplied off the main. But he does not let this high temperature water up into the building mains because if he

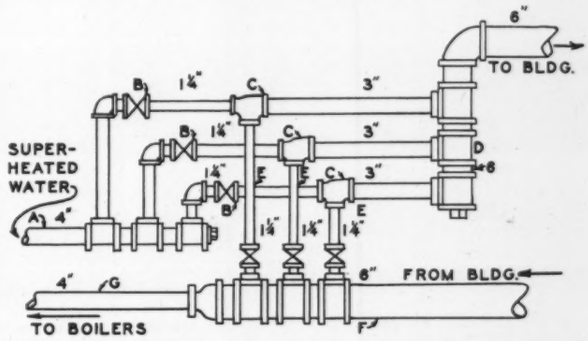


Fig. 2—Detail of Injector Connections in Basement of Each Building

does, of course, the water in the building risers, when it gets near the top and the head is relieved, will start to boil.

To avoid this difficulty he makes a connection to the building supply line through injector fittings and uses part of his building return water to feed the injector suction and get a volume of mixed water beyond his injector which water is below the boiling point and is thus perfectly safe to use in the lines in the building.

As generally carried out this connection is arranged somewhat as shown in Fig. 2. Here the superheated hot water line comes in at *A* and is 4 in. in size; this is then split into three 1¼-in. branches which are carried up and turned horizontally into the injector fittings *C* through gate valves *B*. Beyond the injectors the lines are increased to 3-in. size and are run over and connected into the vertical 6-in. riser *D* which supplies the building. The injector suction pipes *E* are also made 1¼-in. and are connected into the 6-in. return from the building